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Title:
The characterisation and optimisation of earthquake shaking table performance

DEPARTMENT OF CIVIL ENGINEERING

UNIVERSITY OF BRISTOL

THE CHARACTERISATION AND OPTIMISATION OF EARTHQUAKE SHAKING TABLE PERFORMANCE

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A dissertation submitted to the University of Bristol
in accordance with the requirements for the degree of
Doctor of Philosophy in the Faculty of Engineering

September 1998

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Abstract

Shaking tables have been used for many years to perform dynamic testing of structures. However, the dynamic characteristics of both the tables and structures being tested can have a significant effect on the results obtained from the experimental work. This dissertation describes investigations into the factors affecting shaking table experimentation, in particular comparing the performance of four very different European tables that for the first time have been compared to a common standard. The testing techniques developed during this research are now being used at several shaking table facilities throughout Europe.

The implications of the limitations of shaking tables on any testing programme are discussed, and methodologies for reducing or compensating for the effects of shaking table-specimen interaction are proposed. The investigations have shown that it is very important to understand the dynamic characteristics of a shaking table and to take them into account when planning a test. Small errors between desired and actual table motions may have a significant effect on specimen behaviour, and if these errors are ignored, the test results may be affected by the performance of the table, potentially invalidating the experimental research.

However, this research has also shown that although shaking tables may have very different characteristics, excellent table motions can still be achieved using iterative displacement matching, even if the table performance is very non-linear. Significant table-specimen interaction can also be controlled, provided the operator is aware of the behaviour of the table and understands how the design of the experiment will effect the performance of the table.

This dissertation describes and discusses the results of this entire programme of work and looks at the future of shaking table testing for Earthquake Engineering researchers. It is hoped that this dissertation will be of particular benefit to new researchers in the field of Earthquake Engineering.

Acknowledgements

I would like to thank Dr CA Taylor, my supervisor, for his help and encouragement over the past years whilst I have been working on this PhD dissertation.

I also thank the rest of the Civil Engineering Department at Bristol for their support, in particular R Linham, the Laboratory Manager, and the technical staff in the Department for their help with the experimental work. My special thanks go to D Ward, whose specialist knowledge and experience of the Bristol shaking table in use has been invaluable.

Much of the work outlined in this dissertation was made possible by funding from the European Community as part of its "Programme for Human Capital and Mobility - Access to Large Scale Facilities". This programme gave me the opportunity to visit other shaking table facilities in Europe and to discuss with many other researchers how to make best use of shaking tables. I am indebted to all these people, in particular Prof. P Carydis from the Laboratory for Earthquake Engineering, National Technical University of Athens, Greece, Dr G Franchioni from Structural Dynamic Testing Laboratory, ISMES spa, Seriate, Italy, and Dr R Bairrao from LNEC (National Laboratory for Civil Engineering), Lisbon, Portugal.

Finally, I would like to thank Ove Arup and Partners, who originally gave me the opportunity to start this research.

Author's Declaration

I declare that the work in the dissertation was carried out in accordance with the Regulations of the University of Bristol. The research described in this dissertation is based entirely on independent work conducted between November 1993 and September 1998 under the supervision of Dr C.A. Taylor. The work described and ideas recorded are entirely those of the author, except where acknowledged in the text or by reference.

The work contained in this dissertation has not been submitted previously for any other degree or qualification at this or any other University or examining body, and the views expressed in it are those of the author and not of the University of Bristol.

The following papers and reports are based on the research described herein:

Comparative shaking table studies at the National Technical University of Athens and at Bristol University. PG Carydis, HP Mouzakis, EA Vougioukas, CA Taylor & AJ Crewe. 1994. *10th European Conference on Earthquake Engineering, Vienna, Austria.*

The performance of the shaking table control systems at the National Technical University of Athens, Bristol University and at ISMES, Italy. AJ Crewe, CA Taylor, HP Mouzakis, EA Vougioukas, G Franchioni. 1996. *10th International Seminar on Earthquake Prognostics, Cairo, Egypt.*

Comparative Assessment of Shaking Tables. PG Carydis, HP Mouzakis, EA Vougioukas, AJ Crewe, CA Taylor, G Franchioni, R Bairrao. 1996. *11th World Conference on Earthquake Engineering, Acapulco, Mexico.*

Standardisation of Shaking Tables, AJ Crewe(ed.), *ECOEST/PREC8 Report No.1, LNEC Lisbon, pp. 1-200, Aug. 1997, ISBN 972-49-1719-3.*

Signed: A. J. Crewe.

Date: 25/9/98

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Notation

Chapter 3

δ	Deflection
ε	Strain
f	Frequency
ρ	Mass density
σ	Stress
ν	Poisson's ratio
a	Acceleration
g	Gravitational acceleration
m	Number of independent dimensionless groups
n	Number of the basic parameters
q	Pressure
r	Number of fundamental measures
t	Time
v	Velocity
E	Young's modulus
E/ρ	Specific stiffness
EN	Energy
L	Length
L_m	Length scale of model (i.e. shown by subscript 'm')
L_p	Length scale of prototype (i.e. shown by subscript 'p')
S_L	Ratio of length scale of model to length scale of prototype (i.e. L_m / L_p)
S_E	Ratio of Young's modulus of model to Young's modulus of prototype (i.e. E_m / E_p)
S_ρ	Ratio of density of model to density of prototype (i.e. ρ_m / ρ_p)
M	Mass
Q	Force

Chapter 4

δ	Deflection
π	Pi
f	Natural frequency of the oil column
m	Effective mass of table and specimen
A	Effective cross sectional area of the oil column
K	Bulk modulus of the hydraulic oil
L	Length of the oil column

Chapter 6

Δf	Frequency resolution of FFT
Δt	Sampling time interval
N	Number of samples (must be a power of 2)

Chapter 1

Introduction

1.1 Overview

Engineering was described in 1976 by Dr AR Dykes, the President of the Institution of Structural Engineers, as:

> "The art of modelling materials we do not wholly understand, into shapes we cannot precisely analyse, so as to withstand forces we cannot properly assess, in such a way that the public has no reason to suspect the extent of our ignorance"

Earthquake Engineers are faced with similar problems, but in all their areas of uncertainty they face, if anything, even worse problems than do the majority of practising Civil Engineers. The earthquakes themselves are notoriously unpredictable when it comes to deciding on the forces that a structure must be designed to carry. The behaviour of materials and structures which will, in a large earthquake, be taken close to failure is also extremely complicated. Faced with these difficulties, earthquake engineers have to use whatever techniques they have at their disposal to improve their understanding of structural behaviour in earthquakes in order to minimise the risk to the public.

Post-earthquake reconnaissance would have been the only option available to the very first builders, who could only really learn from their mistakes, and by looking closely at those structures that survived an earthquake and those that did not. The advent of experimental test facilities allowed any basic design rules that were developed up to that point to be validated and improved on. Finally, the use of computers opened up the possibility of modelling complete structures analytically and subjecting these computer models to many possible simulated earthquakes. All these techniques are still vital, and only by combining the lessons from all these methods can earthquake engineers increase their understanding of structural behaviour in earthquakes. Experimental modelling and testing is particularly important as it allows engineers to test their theories about structural behaviour, and to validate and verify the computer models that will subsequently be used to help design the actual structures that are to be built in seismic zones.

There are several different experimental techniques that can be used to test the response of structures to simulated earthquake loading, one of which is the use of a shaking table. A shaking table typically consists of a large, rectangular platform that is driven in up to six degrees of freedom (DOF) by servo-hydraulic or other types of actuators. Test specimens are fixed to the platform and shaken. Shaking tables are used extensively in seismic research, as they provide the means to excite structures in such a way that they are subjected to inertia loads representative of true earthquake ground motions. However, as with any experimental apparatus, it is essential that the user has a clear understanding of the capabilities of the apparatus and understands how the apparatus interacts with the test specimen. No experimental apparatus is perfect or free from undesirable aspects. The more complex the apparatus, the more difficult it is to identify and cater for these aspects. This is particularly so for large shaking tables, which are complex electro-mechanical systems. Although an individual table may be similar to others, it will always have features peculiar to itself, which will affect its performance. The control of such systems is not a trivial problem.

Before this research programme started very little work looked specifically at the way in which shaking tables were used by researchers, and at whether the many different types of shaking table facility around the world were being used effectively. It is believed that this is the first research programme to have looked critically at several different shaking table facilities with the aims of improving our understanding of the limitations of this form of testing and of improving the way in which researchers use these important facilities.

This dissertation is the culmination of several years of research into the performance of shaking tables and their use in the field of earthquake engineering research. It has been written in a way that hopefully will increase its value to new researchers in the field of earthquake research; and for this reason it incorporates and discusses some relatively basic material.

1.2 Research objectives

These investigations into the performance of shaking tables began in 1993 in order to confirm whether several different tables could accurately reproduce the same required motions. This work was essential in order to allow an extensive programme of European research (Commission of the European Communities, 1990) to proceed. This research programme planned to use the results from many different tests in several shaking table laboratories. In order to compare results from each of the facilities, the relative performance of each of the facilities had to be assessed. The "Standardisation of Shaking Tables" project (Crewe, 1997) examined three main areas in the use of shaking tables. These were the Software Review, which covered the validation of software used at each site; the Operations Review, which looked at the operational procedures and Quality Assurance (QA) procedures at each site; and the Performance Review, which investigated the dynamic performance of the shaking tables. The main performance tests started with a detailed set of tests being performed on the shaking table in the Earthquake Engineering Research Centre (EERC) at the University of Bristol, UK. These were followed by similar tests on a shaking table in the Laboratory for Earthquake Engineering (LEE) at the National Technical University of Athens (NTUA), Greece, and slightly modified tests on the tables in the Structural Dynamic Testing Laboratory of ISMES, Seriate, Italy, and at the Laboratorió Nacional De Engenharia Civìl (LNEC), Lisbon, Portugal. For the first time a detailed comparison of several shaking tables was possible. This research has produced much valuable information on the characteristics and performance of shaking tables (Carydis et al. 1994, Carydis et al. 1996, Crewe et al. 1996, Crewe 1997).

The initial tests at the four facilities highlighted the need for a detailed investigation into the effect of inaccuracies in the achieved platform motion on the results of shaking table tests. The tests also identified weaknesses in the control methodology of shaking tables, and the need for improved testing techniques to cope with testing of specimens that have significant dynamic interaction with the shaking table. Finally, the tests highlighted a need for completely new control techniques to deal with real-time control of shaking tables and non-linear specimen response during a test. A second set of more detailed tests focusing on these particular issues was subsequently carried out at Bristol in order to address these problems, to validate several new testing techniques developed over the last few years to cope with non-linear specimen performance, and to confirm the adverse effect of rotational components on specimen behaviour.

The key aims of this PhD research programme can be summarised as follows:

1. To produce a detailed characterisation, to a common specification, of the dynamic performance of several shaking tables both with and without a specimen attached.

2. To develop a systematic methodology for regularly assessing the performance of an earthquake shaking table.

3. To identify the strengths, weaknesses and any necessary enhancements of all the tables and if possible make the required improvements.

4. To examine the ability of the tables to respond accurately to input signals and the efficiency of the control software and mechanical control systems in avoiding unwanted platform motions.

5. To investigate the problem of shaking table-specimen interaction and to determine the implications of a lack of control on any tests that are performed.

6. To examine the shaking table testing techniques used in several laboratories in order to compare their efficiency in performing a standardised test.

7. To decide which testing techniques work most efficiently with the ultimate aim of producing a best practice guide for researchers who wish to perform shaking table tests.

All these key issues were addressed as part of this research, and as a result the personnel at all four laboratories now have a much better understanding of the "best practices" for shaking table testing. The research has also generated much quantitative data on the errors that may occur during a shaking table test, and has resulted in proposals for several new methodologies for use in the design of shaking table tests to minimise these errors (see chapter 6).

1.3 Structure of dissertation

This dissertation deals with a study of the performance of shaking tables and of problems associated with their use in earthquake engineering research. The results of an extensive series of tests into the performance of four shaking tables are described and discussed, as are the results of a series of follow-up investigations on the Bristol table. Finally, guidance is provided for researchers who wish to perform shaking table tests.

Chapter 1 has outlined the basic issues regarding shaking table testing and explained some of the background to the research.

Chapter 2 gives a review of the available literature dealing with the performance, control and use of shaking tables.

Chapter 3 reviews the fundamental issues regarding any dynamic testing, and compares and contrasts several of the techniques available to experimentalists who wish to study the dynamic behaviour of structures. The problem of how to model structures accurately is also covered, and guidelines for the choice of test method are presented.

Chapter 4 examines shaking tables in more detail. The structural and mechanical aspects of various types of shaking table are described along with the various standard control methodologies that can be used to control the motion of shaking tables.

Chapter 5 describes the four shaking tables studied as part of this current research, together with the test programme that was used to evaluate them. The main part of the chapter contains the results of an extensive series of tests performed on the four shaking tables, and a discussion of these results. The chapter concludes by describing several sets of additional tests performed at Bristol that further investigated some of the specific problems raised during the study of the four tables. The potential problems caused by interaction between a shaking table and a specimen are described in greater detail, and the implications of inaccuracies in platform motions are outlined.

Chapter 6 looks at several new testing methods that were developed during this research programme and can be used to improve the accuracy of the platform motion during shaking table tests. A few techniques that may significantly improve the use of shaking tables in the future are also briefly outlined, along with some initial results.

Chapter 7 summarises the key results and general conclusions that have been drawn from this PhD research. The chapter finishes by suggesting possible improvements that can be made to shaking table facilities, to improve their effectiveness in the field of earthquake engineering research.

Chapter 2

Literature Review

2.1 Introduction

The first recorded attempt at recreating earthquake motions in a laboratory took place at the beginning of this century (Rogers, 1906) just after the earthquake in California on the 18th April that year. However, a lack of experimental facilities and analytical techniques meant that very few advances were made until much later in the century, and it was the 1st World Conference on Earthquake Engineering, held in 1956 in Berkeley, California, that brought together many of the people who were beginning research in the field of Earthquake Engineering.

By the late 1960's significant advances were being made in servo-control techniques, analogue and digital hardware and dynamic instrumentation, and these were allowing experimentation into the dynamic behaviour of structures. Following these, and subsequent, technical advances, which allowed researchers to study structural behaviour under simulated earthquake loading, much work has been published on the results of specific tests that have been performed on shaking tables. For example, in 1992, at the 10th World Conference on Earthquake Engineering in Madrid, Spain, there were more than 30 papers (just under 2% of the total number of papers) that detailed results of shaking table tests on concrete, steel and masonry frames, and various other civil engineering structures. By 1996, at the 11th World Conference on Earthquake Engineering in Acapulco, Mexico, the number of papers describing the results of shaking table tests had risen to 41 (just over 2% of the total number of papers). From these figures, and looking back at the quantity of previously published work, it is clear that shaking tables are being used increasingly in earthquake engineering research. However, little work on the actual performance of the tables has been carried out. At the 10th World Conference on Earthquake Engineering there were only two papers, by Clark (1992) and Kusner et al. (1992), looking at the performance of shaking tables and servohydraulic systems respectively. Over the next four years a little more research was carried out in this field, and at the 11th World Conference on Earthquake Engineering papers were read by Carydis et al. (1996), Filiatrault et al. (1996), Ventura et al. (1996), Horiuchi et al. (1996) and Murcek et al. (1996) describing various advances in dynamic testing techniques. However, only the first three of these papers were directly applicable to the performance of shaking tables as compared to other types of facility (§3.4), and one of these (Carydis et al., 1996) is a direct result of the research described in this thesis.

2.2 Developments in the use of shaking tables

Several people have produced lists of the various shaking table facilities around the world. The earliest of these (Chowdhury, 1983) listed all the main facilities in the world, while Kamimura and Nakashima (1983) concentrated on the seismic test facilities in Japan. More recently Duarte (1994) produced a comprehensive list of 54 of the largest tables in the world, including descriptions of the types of facility and the basic performance characteristics of each table. The Earthquake Engineering Research Institute, USA, has also produced a detailed assessment of Earthquake Engineering research and testing capabilities in the United States (EERI, 1995).

2.2.1 Design of shaking tables

The initial design and construction of a shaking table has a significant influence on its actual behaviour and subsequent use for dynamic testing. Many papers have been published outlining the design, construction and subsequent upgrading of shaking table facilities, and the most significant of these papers are highlighted below.

Over the years much has been written on the design and construction of specific shaking table facilities. One of the earliest papers to describe the design of two shaking tables was that by Bouwkamp et al. (1969), who also discussed the research potential of these two facilities. Many of the ideas in this paper were taken further by Clark and Burton (1978) and Aristizabal-Ochoa and Clark (1980), who discussed the general principles that must be taken into account in the design of a shaking table system. In particular they argued that, in order to achieve the desired system response at minimum cost, a complete definition of the requirements of the completed shaking table system is essential. They also stated that a shaking table must be designed as a complete dynamic system, that the different interactions between the various elements of the shaking table will all degrade the overall performance, and that they should be identified as early as possible in the design of the system so that appropriate measures can be taken to counteract them.

A new shaking table in the dynamics laboratory of the National University of Mexico is described by Díaz and Del Valle (1977), who also presented a few preliminary results from performance tests. In particular they mentioned the problem of oil column resonance and the adverse effect it has on table performance. Ohtani et al. (1984) described the design, manufacture and construction of a three-axis table for the National Research Center for Disaster Prevention (NRCDP) in Japan. They specifically highlighted the issues surrounding the manufacture of hydrostatic bearings and servo-valves. Reinhorn and Prawel (1983, 1986) described the development of the shaking table at the State University of New York at Buffalo. In particular, they outlined the design of the platform itself, which was constructed from a prestressed composite sandwich structure consisting of a steel frame and a superimposed prestressed concrete grid with ferrocement faces. The ferrocement was found to be particularly useful as the surface layers for the platform. Its unique characteristics led to a very thin yet stiff facing, which greatly improved the overall dynamic behaviour of the structure. Gucci et al. (1986) outlined a rather unusual type of two-axis table that could test very large specimens with simplified time motion consisting of varying amplitude and frequency sine waves.

Minowa et al. (1991) discussed the upgrading of the large scale shaking table at the National Research Center for Disaster Prevention (NRCDP) that took place in 1988. They outlined the improvements to the mechanics of the table, and also discussed how the dynamic characteristics of the reaction block were improved by increasing the foundation mass from 8700 tonnes to 11700 tonnes. Two other tables that have recently been upgraded are the 10 ft x 10 ft table in the University of British Columbia in Canada, and the 20 ft x 20 ft table in the University of California at Berkeley. Ventura et al. (1996) described the improved performance of the table in the University of British Columbia, resulting from the addition of four extra actuators to the shaking table combined with a new hardware control system. Additional actuators were also added to the table at

Berkeley, resulting in a significant improvement in the overturning capacity and performance of the table (EERC, 1997).

The design and construction of the four shaking tables forming ECOEST is of particular interest with respect to the research on which this thesis is based. The design and construction of the six-axis earthquake simulator at the National Technical University of Athens, that was tested as part of this research, was described Carydis et al. (1982). Blakeborough et al. (1986) described the design of the shaking table, hardware and software control systems at Bristol University. Pereira et al. (1985) discussed the outline design of the large three-axis table at the Laboratorió Nacional De Engenharia Civil (LNEC), Lisbon, Portugal. The aims of shaking table design are presented, along with a description of how they were achieved. The control of the actuators that move the shaking table platform is a central problem, and the merits of analogue or digital control systems and real time or adaptive control algorithms are discussed. A detailed description of the design process of the restraining system for this table was given by Emilio et al. (1986). The numerical model developed to help design the table was discussed further by Emilio et al. (1989), and the paper by Duarte et al. (1994) described the construction of this table in detail.

2.2.2 Shaking table-specimen interaction

Although many papers have been published discussing the design and construction of shaking tables, there have been relatively few that looked specifically at the issue of table-specimen interaction when the tables are subsequently used to test various models.

When shaking table manufacturers and operators produce performance details of the shaking tables that they have built or run, they generally look at the maximum performances of the shaking table with no payload and with static payloads. This is obviously not representative of the usual conditions and the loading that a shaking table is subjected to when some sort of flexible specimen is attached to the platform. Some quite detailed analytical studies of shaking table-specimen interaction have been performed (Clark and Cross, 1984; Clark, 1992), but before this PhD research programme started there had been very little experimental work to determine the extent to which these analytical results held true and, in particular, what techniques could be used to alleviate these problems. Earlier analytical work by Takahashi et al. (1974) also showed that mounting a flexible specimen on a shaking table platform could have a significant effect on the frequency characteristics of the overall system, but they did not find that the ability of the table to reproduce specific time histories would be significantly affected.

The first known experimental results showing shaking table-specimen interaction were recorded by Rea and Penzien (1974). In a paper describing a new two-axis shaking table facility the authors noted that when the shaking table platform was excited in its horizontal axis there was also some pitching motion. Takahashi et al. (1974) also showed that mounting a flexible specimen on a single-axis shaking table platform could have a significant effect on the frequency characteristics of the overall system. However, they did not find that the ability of the single-axis table to reproduce the 1940 N-S El Centro earthquake was significantly affected. More recent studies by Rinawi et al. (1988) looked at the interaction between a single degree-of-freedom (SDOF) model and the pitching motion of a shaking table platform. They measured significant table-specimen interaction, and suggested ways in which the recorded behaviour of the model could be adjusted to take this pitching motion of the platform into account.

Blondet and Esparza (1988) carried out a numerical analysis of the shaking table-structure interaction effects that can occur during seismic simulation tests. They described the development of an analytical model which was verified against experimental measurements of the frequency response of the shaking table at the Catholic University of Peru. This model was then modified by the addition of a single DOF viscoelastic oscillator representing a test specimen on the platform, and used to calculate, in the frequency domain, the stability of the system response and the accuracy with which a defined motion could be reproduced on the platform. The authors concluded that the hardware controller in a shaking table cannot eliminate the effects of table-specimen interaction. They noted that the maximum table-specimen interaction occurred with an attenuation of platform response at the natural frequency of the test specimen, and that this is particularly inconvenient as the object of seismic tests is to excite specimens at their natural frequencies in order to cause damage. They also showed that the larger the ratio of the mass of the specimen to the mass of the platform, the more significant the interaction becomes.

While a few other experimental studies have been performed on specific tables since 1988, mainly concentrating on the actual control of the tables, the effects of table-specimen interaction on the performance of shaking tables and the effects of such interaction on test results have been largely ignored until now. Previous to this research, while table-specimen interaction was acknowledged to occur, no attempt had been made to quantify the scale of the problem or to decide whether this interaction need be taken into account in the design of an experiment. Also, because all previous work had been performed only on individual tables, there had been no study of how the differing dynamic characteristics of shaking tables could affect the results of theoretically identical tests. Finally, there had been no attempt by groups from several shaking table laboratories to share software, ideas and testing techniques that could be used to improve the actual dynamic performance of their tables during testing.

2.2.3 Control of shaking tables

The research outlined in the previous section highlighted the problem of table-specimen interaction, but did not give guidance as to how these interaction effects could be minimised or compensated for by shaking table control systems. Nor had any of the researchers provided guidance as to how the errors in platform motion produced by table-specimen interaction could be controlled. These problems have now been tackled by a few researchers, including personnel from the major shaking table manufacturers, and over the years techniques used to control shaking tables have become steadily more advanced.

Abedihayati and Auslander (1977) described and compared two of the earliest techniques that were used to control the response of a single-actuator shaking table. Using simulations, they compared the effectiveness of two algorithms on the performance of a table. Simova and Mamuçevski (1980) compared the motion of a single-axis shaking table platform resulting from the control of either the acceleration or displacement response of the platform. They concluded that actively controlling platform displacements produced the best platform response. This conclusion is re-iterated in a paper by Jujukovski and Mamuçevski (1986), who described the design and outlined the performance of a new two-axis table in Skopje, Yugoslavia. Flesch (1986) and Thewalt et al. (1986) described in detail the numerical methods that could be used to match iteratively the displacement and acceleration earthquake motions on shaking table platforms at that time.

Various other iterative matching techniques have been developed over the years, and Matsuura et al. (1989) discussed the use of power spectrum control techniques to control the motion of a single degree-of-freedom shaking table. They showed that good control performance could be obtained by averaging power spectra measurements. However, they noted that non-linear friction forces in the actuators, and the transient response of specimens with low damping, can cause difficulties with this control method. The paper by Penn et al. (1991) outlined the principles behind a new software package for shaking table testing.

Kusner et al. (1992) looked at the way in which the performance of different hydraulic and mechanical components of a shaking table affect the performance and control of the whole system. They concluded that the errors that occur in servohydraulic systems should be taken into account when operating such a system, although they did not suggest how this should be done.

Some of the most recent work by Filiatrault et al. (1996) critically compared five different techniques that were used to test the performance of a single-axis shaking table with a two degree-of-freedom model mounted on it. This testing is very similar to the work performed as part of this present research, but because the table tested was only a single-axis table many of the issues discovered during this research did not apply. Their results showed that off-line iterative approaches could not completely control a single-axis table, and they concluded that the development of real-time control techniques will be necessary to control the non-linear interaction between table and specimen.

With the increasing use of shaking tables there developed more concern as to the effectiveness of such facilities, particularly for pan-European research, and in 1990 the Commission of the European Communities produced a long term plan for the use of large shaking table facilities in Europe. The European Consortium of Earthquake Shaking Tables (ECOEST) was established soon after this report, and the structure and activities of ECOEST are described by Severn (1994). One of the key pieces of research performed with the co-operation of the ECOEST laboratories was the characterisation of their four large shaking tables, and investigations into the most effective testing techniques for shaking table testing. This research, described in this thesis, also resulted in many publications highlighting the potential problems with the control of shaking tables and some of the solutions to them. The results from the characterisation tests performed in Athens and Bristol were reported by Crewe and Taylor (1994) and Carydis et al. (1994). The information obtained from the subsequent tests at ISMES and LNEC is reported by ISMES (1996) and LNEC (1996). The key findings that were drawn from this data were then discussed by Carydis et al. (1996) and Crewe et al. (1996). A final report, drawing together the data obtained at the four sites, was produced by Crewe (1997), and considered the general issues of shaking table control along with some of the best practices for controlling table-specimen interaction during testing. This report also looked at certain improvements that could be made to the testing techniques used at the four laboratories involved.

2.2.4 Experimental aspects of shaking table testing

The general use of shaking tables and other types of facility for dynamics research has been discussed by many researchers, such as by Popov (1986) and Pereira et al. (1985). These papers described how experimental tests can be used as an aid to understanding structural behaviour in earthquakes, and how shaking tables, in particular, can be used for the seismic testing of models. However, much less work has been published specifically describing how various types of test should be performed, and examining the production of accurate scale models for shaking table testing.

Oberti and Lauletta (1960) wrote one of the earliest papers that deals with the derivation of the similitude relationships (Buckingham, 1914) specifically with respect to earthquake engineering testing; and in particular they looked at some mass concrete mixes that were used to reproduce prototype properties of dams at reduced scales. They described several sets of tests of these scale models of dams on the two-axis shaking table that was installed in ISMES at that time. This shaking table has since been superseded by the much larger six-axis table that was tested as part of this research programme. This work by Oberti and Lauletta (1960) concentrated on the production of suitable scaled materials, and they noted that production of such materials can be extremely difficult. Krawinkler (1979) later described ways of avoiding these problems by the use of artificial mass simulation, a technique that is now used in many shaking table tests to allow the use of prototype materials in the model structure. One of the most extensive discussions of all these issues can be found in 'Structural Modelling and Experimental Techniques' (Sabnis et al., 1983) and anyone who is planning to perform model tests on a shaking table, particularly using modelled materials, should refer to this book which contains many detailed designs for model concretes, masonry and steels.

Another key aspect of any experimental shaking table testing is the choice of the actual motion that the model will be subjected to, and the effect of this decision on the effectiveness of the actual test. Calvi and Kingsley (1996), whilst concentrating on the problems associated with pseudodynamic testing, outlined the critical issue of the choice of input motion for a dynamic shaking table test, and presented results showing the significant differences in non-linear structural response that can be caused by a change in the form of the time-history. In particular, they showed that if the applied time-history generates a large displacement cycle in the structure, then its location in the time-history can radically affect the response of the structure, the final state of damage, and the residual load capacity. They concluded that every test should be analysed carefully, particularly with respect to the expected structural response and the objectives of the experiment.

2.2.5 Specific examples of table use

Shaking tables have been used to perform many different and varied tests since the late 1960's. A comprehensive list of the types of test that have been performed is beyond the scope of this thesis, but a few examples of them are listed below:

- Concrete shear walls and infill frames (Kwan and Xia, 1995).
- Reinforced concrete frames (Dambrisi and Filippou, 1997).
- Shaking table tests of rigid, semi-rigid, and flexible steel frames (Nader and Astanehasl, 1996).
- Dynamic response of steel frames (Nakamura and Wakabayashi, 1986).
- Single-story masonry houses (Clough et al., 1990).
- Simple masonry buildings (Benedetti et al., 1998).
- Retrofit evaluation of reinforced concrete structures (Bracci et al., 1997).
- Strengthening systems for non-seismically designed frames (D'Anzi et al., 1995).
- Seismic behaviour of foundations (Taylor and Crewe, 1996).
- Performance of retaining walls (Crewe et al., 1998).

- Failure of dams in earthquakes (Mir and Taylor, 1995).
- Behaviour of cable-stayed bridges (Garevski and Severn, 1993).
- Guyed towers (Wahba et al., 1997).
- Simulation of pounding between adjacent buildings (Papadrakakis and Mouzakis, 1995).

Shaking tables are also often used to test equipment and instruments that are to be located in nuclear installations. Fischer (1977) gave an overview of the various analytical and testing procedures employed to obtain Nuclear Regulatory licensing for such items, and outlined the need for research into the conservatism of some of the procedures. Stoessel et al. (1983) also commented on these tests, and Taylor et al. (1991) discussed the effect on seismic qualification procedures of the performance of the shaking table being used. Abell et al. (1995) further discussed the conservatism of the most recent procedures (IEEE, 1987), particularly with respect to the regions of the world where there is low seismicity.

Shaking tables are currently being employed in connection with various problems in an increasing variety of situations. In view of this, further research on the reliability and capability of tables will be important, and this could be reflected in the papers appearing in the near future.

Chapter 3

Physical Modelling of Dynamic Behaviour

3.1 Introduction

Experimental modelling and testing is important because it allows engineers to test their theories about structural behaviour, and to validate and verify the computer models that will subsequently be used to design structures built in seismic zones (Popov, 1986). There are many varied forms of dynamic loading that structures are subjected to, such as wind, traffic and earthquake excitation. In order to investigate the way these forces affect the behaviour of structures, many different types of controlled test may be possible where the effect of changes in the various parameters that control the structural behaviour can be individually studied. These tests are particularly important in the field of Earthquake Engineering where the complex inelastic behaviour of members and connections subjected to irregular and unpredictable earthquakes, especially the extreme design events, must be determined. It is also important to obtain repeatability in any experimental work and this is particularly difficult because of the lack of control over material properties, precision of construction (to some extent), and environmental conditions in full-scale structures. Therefore earthquake engineering researchers will always rely on well controlled laboratory experiments. This chapter reviews the methods available for simulating the effects of seismic excitation on models of buildings and other civil engineering structures, as well as on in-situ structures.

Many specific types of experimental facility have been developed over the years that can reproduce, to some limited extent, the various types of dynamic loads that are experienced by real structures. However, each of these types of facility (wind tunnels, external exciters, shaking tables, reaction walls etc.) have their own particular characteristics and restrictions on the types of test that can be performed. The main restriction is usually that they cannot test full-sized structures. An ideal dynamic test would be performed on a full-scale model made using the same materials and construction methods as the prototype (or full-sized) structure, and the exciting forces would be as realistic as possible. Unfortunately, performing full-scale tests is, in all but a few cases, either impractical or impossible. The Cardington test facility (figure 3.1) is exceptional in that it provides a very large enclosed space where full-sized structures can be tested. However, while this facility allows the construction of full-sized structures, only a very restricted range of dynamic loading can be applied to them, and actual earthquake loading (i.e. base excitation) is not possible. If researchers are interested in earthquake effects on structures then they face significant practical problems if they want to perform full-scale tests. For example, in order to test a large multi-storey building under earthquake loading, there would first be the cost of building a real structure and then the cost of providing a method of exciting the structure. This would require a very large and expensive test facility, and while a few such massive test facilities do exist in Japan (figure 3.2) they are very expensive both to build and to run. However, if a small model of the full-sized structure could be produced that behaves in the same way as the prototype (or full-sized) structure, then such a large scale test would become unnecessary. By studying the behaviour of many smaller and cheaper models, researchers could learn much about the behaviour of the full-sized structures. Fortunately it is possible to produce such small scale models, although it may not be possible accurately to replicate every property of the prototype structure at a reduced scale. Dynamic testing can therefore be seen as a compromise between the size and expense of full-scale testing, which avoids the difficulties that are associated with scale models, and testing small, cheap, scaled models that may not be completely representative of the full-sized structures.

When planning research that will investigate, and hopefully improve our understanding of, material or structural behaviour under dynamic loading, it is essential that the researcher is aware of the advantages and disadvantages of testing differently-scaled models and of the effects of the various restrictions imposed by specific test facilities; the test method finally chosen can then be the most appropriate for achieving the specific aims of the research. Ideally, the planned research should also incorporate some field tests and analytical work so that the problem in question can be studied from as many angles as possible. In this way it is easier to compensate for the disadvantages in each technique used. The next two sections (§3.2 and §3.3) deal specifically with the problems of producing accurate scale models of full-sized prototype structures. These are followed by descriptions of several different types of test facility that are available to researchers studying the response of structures to earthquake loading.

3.2 Model testing theory

The difficulties involved in performing full-scale tests means that the majority of practical tests rely on the production of small scale test models. These are often not easy to produce; the specific properties of any model will depend on the type of test that is proposed and on whether it is essential to produce a completely accurate model. One method for determining the various scaling factors for the individual properties of a model is described below, and is based on the work of Krawinkler (1979), Sabnis et al. (1983), Oberti and Lauletta (1960).

3.2.1 Derivation of model scaling factors

1) Make a list of the minimum number of physical parameters that are directly relevant to the problem. For many structural dynamic problems such a list is likely to include Length (L), Time (t), Gravitational Acceleration (g), Acceleration (a), Mass Density (ρ), Stress (σ), Young's Modulus (E) and Deflection (δ). These parameters are shown highlighted (*) in column (1) of table 3.1. There may be other parameters that can be derived from these basic parameters, and some of these that are relevant to dynamic testing are also shown in column (1) of table 3.1.

   Then define 'n' as the number of the basic parameters that exist for the problem, in this case n = 8, and 'r' as the number of fundamental measures that are used to form these parameters, in this case length, time and mass, so that r = 3.

2) Buckingham's Pi Theorem (Buckingham, 1914) of dimensional analysis states that "*any dimensionally homogeneous equation involving certain physical quantities can be reduced to an equivalent equation involving a complete set of dimensionless products*". This means that this list of 'n' basic parameters, some function of which defines the problem, can be converted into an equivalent set of independent dimensionless groups of these parameters, which also define the problem. There will be 'm' = n - r independent dimensionless groups that can be formed. In this example, m = 8 - 3 = 5.

Table 3.1 Scaling requirements for various types of model test.

Parameter to be scaled	Dims.	Scale factors for particular types of model					
		True replica models		Shaking table models			
		Normal gravity	Artificial gravity (centrifuge models)	Artificial mass simulation (material modelled)	Artificial mass simulation (prototype material)	Gravity forces neglected (material modelled)	Gravity forces neglected (prototype material)
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Length (L) *	L	$\underline{S_L}$	$\underline{S_L}$	$\underline{S_L}$	$\underline{S_L}$	$\underline{S_L}$	$\underline{S_L}$
Time (t) *	T	$S_L^{1/2}$	S_L	$S_L^{1/2}$	$S_L^{1/2}$	$S_L S_E^{-1/2} S_\rho^{1/2}$	S_L
Frequency (f)	T^{-1}	$S_L^{-1/2}$	S_L^{-1}	$S_L^{-1/2}$	$S_L^{-1/2}$	$S_L^{-1} S_E^{1/2} S_\rho^{-1/2}$	S_L^{-1}
Velocity (v)	LT^{-1}	$S_L^{1/2}$	$S_L^{1/2}$	$S_L^{1/2}$	$S_L^{1/2}$	$S_E^{1/2} S_\rho^{-1/2}$	1
Gravitational Accel. (g) *	LT^{-2}	$\underline{1}$	S_L^{-1}	$\underline{1}$	$\underline{1}$	neglected	neglected
Acceleration (a) *	LT^{-2}	1	S_L^{-1}	1	1	$S_L^{-1} S_E S_\rho^{-1}$	S_L^{-1}
Mass Density (ρ) *	ML^{-3}	S_E / S_L	$\underline{1}$	S_E / S_L **	$1 / S_L$ **	$\underline{S_\rho}$	$\underline{1}$
Strain (ε)	-	1	1	1	1	1	1
Stress (σ) *	$ML^{-1}T^{-2}$	S_E	1	S_E	1	S_E	1
Young's Modulus (E) *	$ML^{-1}T^{-2}$	$\underline{S_E}$	$\underline{1}$	$\underline{S_E}$	$\underline{1}$	$\underline{S_E}$	$\underline{1}$
Specific Stiffness (E/ρ)	L^2T^{-2}	S_L	1	S_L	S_L	$S_E S_\rho^{-1}$	1
Deflection (δ) *	L	S_L	S_L	S_L	S_L	S_L	S_L
Force (Q)	MLT^{-2}	$S_E S_L^2$	S_L^2	$S_E S_L^2$	S_L^2	$S_E S_L^2$	S_L^2
Energy (EN)	ML^2T^{-2}	$S_E S_L^3$	S_L^3	$S_E S_L^3$	S_L^3	$S_E S_L^3$	S_L^3
Pressure (q)	$ML^{-1}T^{-2}$	S_E	1	S_E	1	S_E	1
Mass (M)	M	$S_E S_L^2$	S_L^3	$S_E S_L^2$ ***	S_L^2 ***	$S_E S_L^3$	S_L^3
Poisson's Ratio (ν)	-	1	1	1	1	1	1

* The basic physical parameters that are used to define the dynamic behaviour of structures

** Use these parameters when considering distributed mass systems

*** Use these parameters when considering lumped mass systems

There may be several different, equally valid, sets of 'm' dimensionless groups that can be formed. One such set of 5 dimensionless groups for the 8 parameters above is:

$$\frac{\sigma}{E}\ ,\ \frac{\delta}{L}\ ,\ \frac{a}{g}\ ,\ \frac{gL\rho}{E}\ ,\ \frac{t}{L}\sqrt{\frac{E}{\rho}} \tag{3.1}$$

Several other possible sets of dimensionless groups exist for these 8 parameters. These could be found by trial and error or by other numerical means (Sabnis et al., 1983).

3) Ideally, for a completely accurate model, the value of each of these dimensionless groups in the prototype (shown as subscript p) and the model (shown as subscript m) would be the same, e.g. $\frac{\sigma_p}{E_p} = \frac{\sigma_m}{E_m}$. These equations can be simplified if we define a set of scale factors, such as S_E, which would be the ratio of the Young's Modulus in the model (E_m) to the Young's Modulus in the prototype (E_p); that is, $S_E = E_m / E_p$. In a similar way, each of the dimensionless groups in eqn. 3.1 can be rewritten in terms of the ratios of scales of the basic parameters:

$$\frac{S_\sigma}{S_E}\ ,\ \frac{S_\delta}{S_L}\ ,\ \frac{S_a}{S_g}\ ,\ \frac{S_g S_L S_\rho}{S_E}\ ,\ \frac{S_t}{S_L}\sqrt{\frac{S_E}{S_\rho}} \tag{3.2}$$

Thus if we want a completely accurate model then all of these scale ratios should be equal to unity i.e.

$$\frac{S_\sigma}{S_E}=1\ ,\ \frac{S_\delta}{S_L}=1\ ,\ \frac{S_a}{S_g}=1\ ,\ \frac{S_g S_L S_\rho}{S_E}=1\ ,\ \frac{S_t}{S_L}\sqrt{\frac{S_E}{S_\rho}}=1 \tag{3.3}$$

(It should be noted that Sabnis et al. (1983) and Oberti and Lauletta (1960) define the scale factors as the ratios of the properties in the prototype to the properties in the model; i.e. $S_E = E_p / E_m$ rather than as shown above).

4) If there are 'r' fundamental measures, then any values can be chosen for 'r' of the scale factors. These factors are then used to derive the scale factors for the rest of the parameters. The scales for the rest of the parameters are deduced from the dimensionless groups (eqn. 3.3). For dynamic problems it is generally logical to use

length (L), gravitational acceleration (g) and Young's modulus (E) as the three properties that will be scaled, because we have either complete control or no control at all over these parameters. For normal laboratory testing the model scale length will be chosen as desired, therefore S_L can take any value. However gravitational acceleration cannot be changed (unless we are performing the test in a centrifuge, see §3.2.2), therefore $S_g = 1$. By looking at the fourth part of eqn. 3.3, $\left(\frac{S_g S_L S_\rho}{S_E} = 1\right)$ and setting $S_g = 1$ we can see that:

$$S_\rho = \frac{S_E}{S_L} \tag{3.4}$$

This means that it is impossible to make an accurate model in a 1g environment using the prototype materials, as it is impossible to have both $S_\rho = 1$ and $S_E = 1$ and still have eqn. 3.4 holding true for any value of S_L. Therefore, either the material has to be scaled, which is often very difficult, or other alternative methods of modelling that allow us to work with prototype materials while still obeying all the scaling laws must be found. The difficulty with modelling materials can be seen by looking at eqn. 3.4, which shows that if the Young's modulus remains constant then the material density has to increase as the model becomes smaller, and that if the density is kept constant then the Young's modulus must decrease as the model gets smaller. It is hard to find such model materials that have the correct properties at any reasonable model scale i.e. denser but less stiff than the prototype material. If it is possible to find suitable model materials such that $S_\rho = \frac{S_E}{S_L}$ then it will be possible to produce an accurate scale model in a 1g environment, so long as all the remaining parameters are scaled in accordance with the remainder of eqn. 3.3. The complete set of scaling requirements derived from eqn. 3.3 for true models in a 1g environment are shown in column (3) of table 3.1. The three scale factors (S_L, $S_g = 1$, and S_E) used to derive all the rest are shown in bold and underlined.

3.2.2 Scale models for centrifuge testing

If it necessary for the model to be made from the prototype material because suitable model materials do not exist, then there are some options available to researchers. The most obvious way to improve matters would be to allow scaling of gravity, and this can be achieved in a centrifuge. Then it is possible to choose S_L, S_ρ and S_E as the three basic parameters that will be scaled. Setting $S_E = 1$ and $S_\rho = 1$ (i.e. using the prototype material) in the fourth part of eqn. 3.3, $\left(\frac{S_g S_L S_\rho}{S_E} = 1\right)$ we can see that:

$$S_g = \frac{1}{S_L} \tag{3.5}$$

This effectively means that a model that has a geometry that is $\frac{1}{5}$ of the prototype, for example, should be tested at 5g to reproduce the correct prototype stresses. The other scaling factors that are derived from eqn. 3.3 for centrifuge testing with prototype materials are shown in column (4) of table 3.1. The three scale factors (S_L, $S_\rho = 1$, and $S_E = 1$) used to derive all the rest are shown in bold and underlined.

3.2.3 Shaking table models with artificial mass

An alternative method of simulating gravitational effects that can be used in a 1g environment is to increase the effective density of the structure (Krawinkler, 1979). If we choose S_L, S_g and S_E as the three basic parameters that will be scaled, and set $S_E = 1$ and $S_g = 1$ (i.e. using the prototype material and not increasing gravity), in the fourth part of eqn. 3.3, $\left(\frac{S_g S_L S_\rho}{S_E} = 1\right)$ we can see that:

$$S_\rho = \frac{1}{S_L} \tag{3.6}$$

i.e., if we are modelling at $\frac{1}{2}$ scale we need material that is twice as dense. However, rather than attempting to do this directly, it may be possible simply to add additional mass to the structure to increase the effective density, as long as this additional mass has no other effect on the structure, e.g. does not also contribute to an increase in structural stiffness. If the system is being modelled as a lumped mass system, this can be simply achieved by the addition of extra mass at the relevant points according to the relationship of the model-to-prototype ratio, i.e. $S_M = S_L^2$, as marked with *** in column (6) of table 3.1. If prototype materials are not being used for the model, then $S_M = S_E S_L^2$ (column (5) of table 3.1). In a system with distributed mass this addition of mass is harder to achieve, but it is still often possible. In this case distributed mass should be added so that the combined mass density of the model and the additional mass obeys the rule $S_\rho = \frac{S_E}{S_L}$, as marked with ** in column (5) of table 3.1 for scaled materials, and $S_\rho = \frac{1}{S_L}$, as marked with ** in column (6) of table 3.1 for prototype materials. For example, if a model of a lumped mass system is made using prototype materials with a length scale $\frac{1}{3}$ of the prototype, the lumped masses on the model should be $\frac{1}{9}$ of those the on prototype rather than $\frac{1}{27}$ as might be expected if based on the cube of the scale length. An equivalent procedure has to be applied to if we are considering a distributed mass system, although the actual mass of the model can be deducted from the additional mass that needs to be added to the elements of the model. Two sets of scaling factors that are derived from eqn. 3.3 for shaking table testing with modelled and prototype materials are shown in columns (5) and (6) of table 3.1. The three scale factors (S_L, S_ρ, and S_E) used to derive all the rest are shown in bold and underlined. When the materials are being modelled S_ρ and S_E can take any value, and when prototype materials are being used $S_\rho = 1$ and $S_E = 1$.

3.2.4 Shaking table models with gravity forces ignored

If the problem being investigated is such that gravitational effects can be ignored, for example if the model is intended to be linear and elastic, then the gravitational effects can normally be treated separately from the dynamic effects and can therefore be ignored in the derivation of the scaling rules. Therefore $\frac{S_a}{S_g}$ need not be 1, and this part of eqn. 3.3 is not used to derive any other scales; in addition S_g in the fourth part of eqn. 3.3, $\left(\frac{S_g S_L S_\rho}{S_E} = 1\right)$ can be replaced by S_a. Two sets of scaling factors that are derived from eqn. 3.3 for shaking table testing with modelled and prototype materials are shown in columns (7) and (8) of table 3.1. The three scale factors (S_L, S_ρ, and S_E) used to derive all the remainder are shown in bold and underlined. When the materials are being modelled, S_ρ and S_E can take any value, and when prototype materials are being used, $S_\rho = 1$ and $S_E = 1$. Unfortunately, because the behaviour of most engineering materials is dependant on compressive loading as well as shear and flexural loading, this technique is not normally appropriate for seismic testing.

3.3 Potential errors

There are many potential sources of error that can occur in the design and construction of a test model. In §3.2 the many conceptual difficulties in creating accurate small scale models of prototype structures were outlined. The main difficulty arises from the problems of obtaining materials with the correct scaled model properties (§3.2.1 part 4). This issue becomes even more difficult when the non-linear behaviour of the model material has to be modelled accurately or when composite materials, like concrete, need to be modelled. In either of these cases researchers need to look very carefully at the model materials being proposed for a test to decide whether any differences between the prototype and model material behaviour are significant. Sabnis et al. (1983) give many specific examples of differently scaled material models of steel, concrete and masonry, and discuss how the specific requirements of a particular test might be met. However, if perfect model materials cannot be obtained, as is generally the case, then there is the potential for large errors in the model finally created. Even if a material with approximately the correct properties can be obtained, there will still be errors between the actual and theoretically desired properties that may affect the overall behaviour of the model during testing. Other problems may occur when a model using prototype materials is loaded with additional mass to achieve the correct overall scale properties (§3.2.3). For example, the way in which the additional mass is attached to the model must be carefully designed if the masses themselves are not to contribute to the stiffness of the model.

Additional problems are also likely to occur if very small scale models are to be tested. For example, smaller models will experience smaller deflections under the applied loads, and these will become much harder to measure accurately as the scale becomes smaller. The strain distributions in the model will also occur over smaller lengths and will require smaller, more accurate strain gauges in order to measure the desired point strains rather than the averaged strains which would be measured by a larger strain gauge on the model. There may also be other local effects that become important if very small scale models are constructed. For example, if the model is welded then heat effects from the welding process, which can probably be ignored in the full-scale structure, may have a significant effect on the model. On the other hand, larger models are more likely to interact with the system that is being used to test them (§5.4.4), so a compromise will be needed between the errors that will occur at these two extremes of model scale.

It is important that researchers are aware that there will always be some errors even in the best designed test. No test can be perfect, and various assumptions, either explicit or implicit, will have been made in the design of the test programme. Therefore, while a test can provide very useful information, experimental research should nevertheless be seen as just one of the tools that is available to help to improve our understanding of structural behaviour. However, a combination of the various types of model testing (§3.4), full-scale field testing (§3.4.5) and analysis offers a powerful approach to extending understanding of the seismic behaviour of structures and systems. Also, if careful consideration is given to all the possible errors that may occur in a model test, many of these can be designed out – and it should then be possible to conduct experimental research that can make significant advances in the field of earthquake engineering.

3.4 Dynamic testing techniques for earthquake excitation

There are several testing techniques that can be used to investigate the dynamic behaviour of structures. However, each system has advantages and disadvantages and no system will be ideal in all circumstances. The next sections outline and give examples of five of the basic methods available to earthquake engineers and researchers who wish to study the behaviour of structures which are subjected to earthquakes. The first of these is the pseudo-static method of testing (§3.4.1) where a specimen is subjected to previously selected cyclic forces or displacements. The second procedure employs a centrifuge (§3.4.2) with additional tangential excitation to apply an additional dynamic motion representative of an earthquake to the specimen being tested. The third procedure employs a shaking table (§3.4.3) programmed to apply scaled earthquake input based on available accelerograms. The fourth, the pseudo-dynamic method of testing (§3.4.4), combines the pseudo-static procedure with an on-line computer, which enables the application of a random sequence of displacements corresponding to a previously recorded earthquake while also taking into account the continuously changing structural stiffness. The fifth, a non-destructive method of testing of actual structures (§3.4.5), consists of recording and interpreting ambient or forced vibrations of the structure in the elastic regime. The advantages and disadvantages of each system are briefly discussed.

3.4.1 Materials and component testing

Materials and component testing (ASTM, 1983, 1993) is generally one of the smallest and easiest forms of testing. Tests of these kinds can be performed in many different ways, but in all cases a small specimen of material, an actual component or a structural model is subjected to dynamic forces and its behaviour under these forces is studied. Many different test machines exist, and the choice of machine will depend on the size of specimen to be tested and the types of loads that need to be applied. A typical test machine is shown in figure 3.3; this machine is capable of applying cyclic axial compression and tension forces up to 500 kN and a simultaneous torsion about the axial axis of 20 kNm at frequencies of up to 200 Hz. It is also possible to test very small structural models (likely to be scaled at around 1:30) with a machine of this kind. Ensuring that such a small model is representative of an actual structure is difficult, and the inertia forces on the actual structure will not be modelled in such a test. Therefore, while these cyclic test machines can provide very useful information about the local behaviour of materials and components, they cannot tell us how a whole structure is likely to behave. Typically these machines are used to perform fatigue tests of connection details and to test concretes, steel, soils or other materials. The results of these types of tests are essential in the development of accurate analytical models of material behaviour. These numerical models may then be used to try to predict the behaviour of much larger structures formed from the materials tested. However, the behaviour of a complete structure is generally much more complicated than being simply a sum of the local behaviours of all the parts of the structure, and to validate computer models of complete structures it is necessary to model and test such complete structures in such a way that the inertia forces are modelled and any other global behavioural effects can be studied. The three most commonly used techniques for doing this are described below.

3.4.2 Centrifuge testing

A large centrifuge (see figure 3.4) is a device within which structural models, components and soils can be subjected to increased gravity loading. It typically consists of a large arm rotating at high speed with a freely rotating box at the free end containing the specimen. With the centrifuge at Cambridge University (Lee & Schofield, 1989), an earthquake input motion is generated by running a sprung wheel, mounted on the end of the box, along a 'bumpy road' around the centrifuge. More modern centrifuges include hydraulic actuators to provide the necessary seismic excitation. Data from transducers are recorded, and from these data the dynamic behaviour of the specimen can be interpreted.

The simulated gravity on the specimen increases with the square of the angular velocity of the centrifuge. Therefore by adjusting the speed of a centrifuge it is possible to increase gravity to a point where the stress in the model is the same as that in an equivalent point in the prototype. Only by increasing gravity can the self weight of a material be increased to simulate the correct stresses that would be present in the prototype. This technique allows tests to be performed at very reduced scales, while still providing data applicable to full-scale problems. The tests can be performed on any particular soil type and/or deposit, and/or for any structure configuration. However, centrifuges are generally small (capable of carrying specimens up to about 1m x 1m x 1m) and hence cannot accommodate large models. Therefore while gravity scaling means that prototype materials can be used in the scale model while still achieving the true prototype stress relationships, the models themselves may be very difficult to produce at the required very small scales (often as small as 1:300). In addition, the motions that the model can be subjected to are also rather limited (Lee & Schofield, 1989), and only single-axis shakes are possible.

Due to the difficulty in producing very small scale structural models, centrifuge testing is commonly restricted to the testing of geotechnical problems, in particular the study of liquefaction, soil-structure interaction, and slope stability problems under static and dynamic loading. Two examples of successful very-small-scale-modelling of embankment dams can be found in Clough and Pirtz (1956) and Seed and Clough (1963). These papers describe the modelling of 1:150 and 1:300 scale models of dams that were tested, in these cases, using a very simple shaking table, but similar static and dynamic tests on these types of models are often performed in centrifuges. Mir (1994) also discusses the scaling issues associated with the production of very small scale models of concrete gravity dams for testing in centrifuges or on shaking tables.

More detailed discussion of the use of centrifuges for structural and geotechnical modelling is beyond the scope of the dissertation, but further information can be found in Schofield (1980) and Sabnis et al. (1983).

3.4.3 Shaking table testing

An earthquake shaking table (see figure 3.5) is a device for shaking large structural models and components with a wide range of simulated ground motions, including reproductions of recorded earthquakes. It typically consists of a large, rectangular platform that is driven in up to six degrees of freedom (DOF) by servo-hydraulic actuators. Test specimens are fixed to the platform and shaken. Using data recorded from transducers it is possible to interpret the dynamic behaviour of the specimen.

Shaking tables are essential tools in earthquake engineering research because they allow the study of the effects of true inertia forces on test specimens which are much larger than those that can be tested in a centrifuge. However, because gravity cannot be increased, scaling the model's properties becomes much harder. In order to reduce the effect of this scaling, the specimens should be manufactured as near to full scale as is practical. The larger the specimen, the less difficult the scaling issues become (§3.2). Therefore the ideal shaking table would be capable of testing full-scale structures so that scaling does not become an issue. However, for massive structures such as dams, this is clearly impossible. Some very large shaking tables, with the capacity to shake full-scale buildings, do exist; for example, the 15m by 15m table with a 1000 tonne capacity located on Tadotsu Island in Japan. However, facilities such as these are extremely expensive to build ($200m upwards) and their running costs, together with the cost of the production of full-scale models, makes them impractical for most normal research. For this reason, test specimens are normally constructed at model scale and shaken on smaller tables such as the 6 m by 6 m table at Berkeley in California (EERC, 1997) and the 8 m by 8 m table in the Earthquake Disaster Prevention Research Centre at the Public Works Research Institute, Ibaraki-ken, Japan (PWRI, 1997). In some cases this modelling can limit the scope of the tests, as small scale reproduction of the properties of materials like concrete is very difficult. However, it may be possible to add additional mass to the models being tested (figure 3.6) to enable the scaling laws to be obeyed (§3.2.3). With this technique it is then possible to perform detailed tests on small models constructed with the prototype materials in a 1g environment. Figure 3.6 shows a large 2-Dimensional (2-D) composite steel frame being subjected to an in-plane horizontal simulated earthquake. Figure 3.7 shows a smaller 3-Dimensional (3-D) concrete model, again with additional added masses, that is being subjected to a two-axis horizontal shake.

It is apparent that a testing rig that had a combination of both centrifuge and shaking table (Schofield & Steedman 1988) would solve many of the problems described above. These types of test rig do exist, a typical example being the centrifuge at the Port and Harbour Research Institute (PHRI) which contains a small (400 mm x 180 mm) shaking table (Inatomi et al. 1988). While such facilities do have a place in dynamic testing, they are obviously mechanically and electronically very complicated (much more so than separate centrifuges or shaking tables) and unless a centrifuge can be built with a table at least 2m or 3m square which can accommodate large models and also be controlled accurately, then the larger centrifuges and shaking tables, even with their limitations, will remain essential parts of the toolkit of facilities available to experimentalists.

3.4.4 Pseudo-dynamic (PSD) testing

In some circumstances it may be essential to test at full scale, either because testing at a reduced scale would be too restrictive on the types of material that could be used or because construction of a small model would be too difficult. In these circumstances neither centrifuges nor shaking tables are likely to be appropriate for most structures. However, pseudo-dynamic test rigs, such as the ELSA (European Laboratory for Structural Assessment) reaction wall at the Joint Research Centre (JRC), in Ispra, Italy allow full-scale specimens to be tested, although not with true inertia forces. The following information (which has been included for completeness) is based on material obtained from the internet site* at ELSA in the JRC, Italy.

* http://tina.sti.jrc.it/ELSA/ELSApseudo.html

A reaction wall (see figure 3.8) is a large, very stiff wall that is used to provide support for actuators that are then used to deform large or full-size structures. Such a facility can be used to perform static or cyclic tests of large structures, but it is also possible to use a technique called pseudo-dynamic testing to simulate earthquake loading of full-scale structures. The reaction wall facility shown in figure 3.8 (which is at ELSA in the JRC, Italy) is one of the largest reaction walls in the world. This 16m wall is designed to resist forces of up to 200 MNm at its base, and can easily carry the forces necessary to deform and damage full-scale structures (Donea et al., 1996).

A pseudo-dynamic test is one which, although carried out quasi-statically, uses on-line computer calculation and control, together with experimental measurement of the actual properties of the structure, to provide a realistic simulation of the dynamic response. The equations of motion for a discrete parameter model of the test structure are solved on-line using a step-by-step numerical integration method. Inertial and viscous damping forces are modelled analytically – a relatively straightforward matter compared to the non-linear structural restoring forces, which are measured experimentally because of the virtual impossibility of modelling them accurately. The process automatically accounts for the hysteretic damping due to inelastic deformation and damage of the structural materials which is the major source of energy dissipation.

When the PSD test is applied to an earthquake simulation test of a civil engineering structure, a record of an actual or artificially generated earthquake ground acceleration history is given as input data to the computer. The horizontal displacements of the floors (the levels at which the mass of the building can be considered to be concentrated) are calculated for a small time step. These displacements are then applied to the structure by servo-controlled hydraulic actuators fixed to the reaction wall. Load-cells on the actuators measure the forces necessary to achieve the required deformation (the structural restoring forces) and these are then used in the next step of the calculation (Seible et al., 1996).

Since the inertia forces are modelled there is no need to perform the test on the real time-scale and typically an earthquake lasting some ten seconds in real time is simulated in a pseudo-dynamic test in about an hour (Mahin and Shing, 1985). This is one of the major advantages of the method – it is possible to test very large models with only a modest hydraulic power requirement. On the other hand the more conventional shaking table tests, although made in real time, are restricted to components or small-scale models of large structures. The second major advantage over a shaking table is the possibility to monitor very closely the progression of damage in the structure, and to stop at any moment for a detailed examination or to prevent complete collapse. The two test methods are in fact complementary – shaking tables being used for preliminary tests and parameter studies at small scale, or when materials which have properties which are very rate-dependent are involved, or when structures with fully distributed mass are concerned. Pseudo-dynamic tests are especially useful for confirmatory tests at full scale where the exact material and construction details can be reproduced, or when multi-point input loading is required.

By using a mathematical technique known as sub-structuring, significant further developments are possible. With this procedure only the most interesting part of a structure is tested experimentally, while the rest is modelled analytically (Schneider and Roeder, 1994; Shing et al., 1996). The computer then accounts for the interactions between the two parts of the structure in calculating the displacements to impose on the tested part. Thus, structures much larger than the laboratory itself, such as for example a bridge, can be tested. Here, by assuming elastic behaviour of the bridge deck, the computer can account for the effects of its behaviour in calculating the displacements to impose on the piers, and the physical testing can be limited to the piers alone (figure 3.9).

Alternatively, when rate-dependency of structural materials is very important, somewhat faster testing speeds can be achieved by reducing the physical model to just those few components expected to show non-linear behaviour while the rest which behave linearly are simulated in the computer. Soil-structure effects may also be taken into account by this means provided that a suitable analytical model of the soil behaviour is available.

Using PSD testing it is possible to study the dynamic behaviour of full-sized structures, but only at slow strain rates. The use of substructuring, probably the most important advantage of reaction wall testing, also relies on the use of a computer simulation to calculate the elastic behaviour of the majority of the structure. Therefore, if the material behaviour is suspected to be strain rate dependant or if large parts of the structure cannot be assumed to behave linearly, then PSD may not be an appropriate method of performing that particular test.

3.4.5 Full-scale field testing

One final option for studying the dynamic behaviour of structures is to excite real structures and record their behaviour under some form of externally applied loading (DTA / NAFEMS, 1993). An example of this technique is shown in figure 3.10, where eccentric mass exciters mounted on the crest of a dam are being used to vibrate the dam and hence determine the natural frequencies and damping parameters of the structure. However, this type of testing is limited by the amount of force that can be injected into the structure, which generally means that the structure can only be excited in its linear range.

3.5 Choosing the type of test to be performed

Table 3.2 lists the main advantages and disadvantages of each of the types of test outlined in section §3.4, along with the typical types of test that are performed using each facility. It is clear that different types of test (materials, centrifuge, shaking table, PSD or full-scale) will be appropriate in different circumstances. Researchers in the field of earthquake engineering should be aware of the need to select the right type of facility for a specific test, depending on the type of research and the type of results that are wanted. If the research requires study of the overall behaviour of a structural system then materials testing and PSD testing will be inappropriate. Study of non-linear behaviour may be difficult, in the case of full scale testing, because of the difficulty of supplying enough energy to the structure to produce a non-linear response. This leaves a choice between centrifuge and shaking table model tests, both of which can reproduce the actual loading that the prototype structure would be subjected to. The choice between these two options will then be dependent on the ease of production of the relevant scale model and the access to an appropriate facility. If, on the other hand, the research only requires investigation into the response of part of structure, then the difficulties associated with the creation of accurate models may be avoided by opting for materials, PSD or full scale testing methods.

Table 3.2 Summary of the main dynamic testing techniques for earthquake excitation.

Testing Method	Typical tests	Advantages	Disadvantages
Materials and component testing	• Small concrete / steel / masonry specimens • Full-scale joints and connections • Fatigue tests	• Only small specimens need be manufactured	• Inertia forces not modelled • Only small tests are possible
Centrifuge testing	• Models of soil embankments • Models of piles and foundations including the soil	• Self weight modelled accurately under the increased gravity • Strain rate effects are modelled	• Specimens at very small scale • Limited input motions available
Shaking table testing	• Models of buildings	• Strain rate effects are modelled • Any input motion can be used	• Material self weight cannot be modelled • Inertia forces can be modelled correctly but only by adding mass
Pseudo-dynamic (PSD) testing	• Full-scale versions of the non-linear elements within a complete structure	• Full-scale testing of whole structures or elements is possible.	• Inertia forces not modelled • Strain rate effects cannot be modelled • Substructuring techniques rely on an accurate computer model
Full-scale field testing	• Modal characterisation of dams, buildings and other structures	• Complete full-scale structures can be tested	• Only low levels of excitation force are possible • May not be possible to test into non-linear range

3.6 Conclusions

This chapter has discussed the difficult issues facing earthquake engineering researchers who wish to perform testing of dynamic behaviour. While full-scale testing might be ideal, it is almost always impossible for financial reasons and vary rarely practical. The production of correctly scaled models for dynamics research can be very difficult, and so most tests will involve some form of compromise between using larger models that are simpler to construct but require more comprehensive test facilities, and smaller models that become progressively harder to scale accurately. The five main experimental techniques that are used in dynamics research have been discussed along with the advantages and disadvantages of each technique. Since shaking table testing offers the possibility of subjecting structures to ground motions such as occur in real earthquakes, such testing will always be important when looking to increase the understanding of the dynamic behaviour of structures. It is therefore important to be able to perform such tests properly. However, it must not be forgotten that effective research uses all the techniques at the researchers disposal and that shaking tables are just one of the tools available. International collaboration of researchers to allow the use of every available method is obviously desirable, and a combination of materials testing, centrifuge testing, shaking table and pseudo-dynamic experiments with field testing and analysis offers a powerful approach to extending understanding of the seismic behaviour of structures and systems.

(Original in colour)

Fig. 3.1 The Cardington test facility, UK

[From 'The Structural Engineer', Vol. 76, No. 14, 21/7/98]

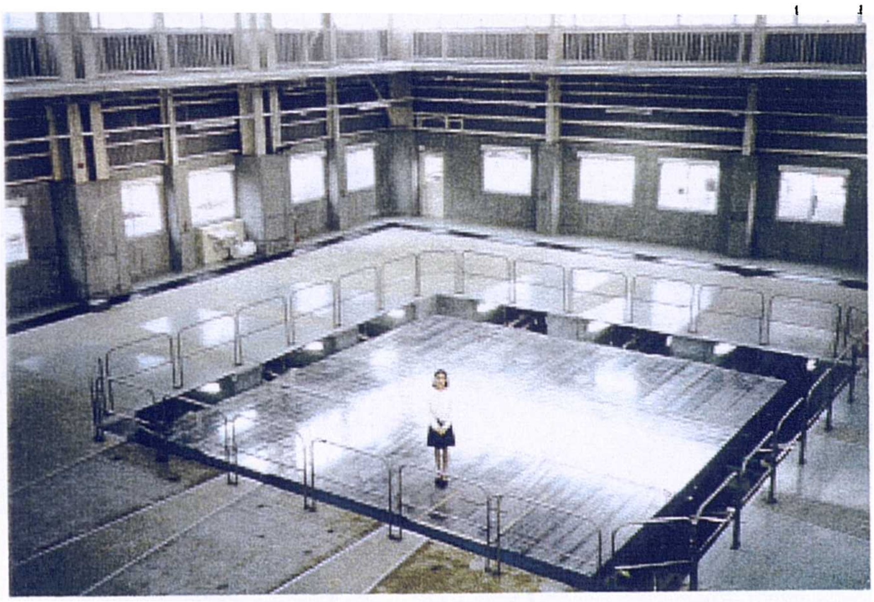


Fig. 3.2 The 8m by 8m table in the Earthquake Disaster Prevention Research Center

[From publicity material, Public Works Research Institute, Ministry of Construction, Japan]

(Original in colour)

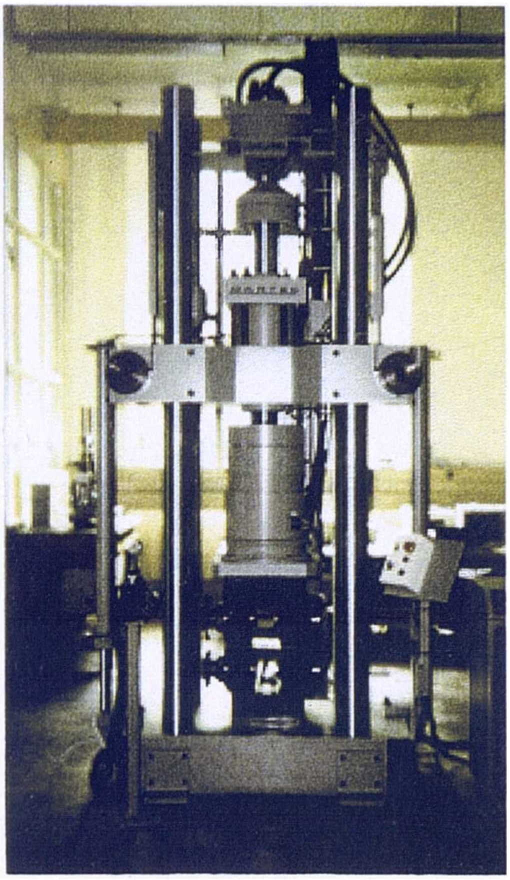


Fig. 3.3 The combined axial and torsion testing machine at Bristol University

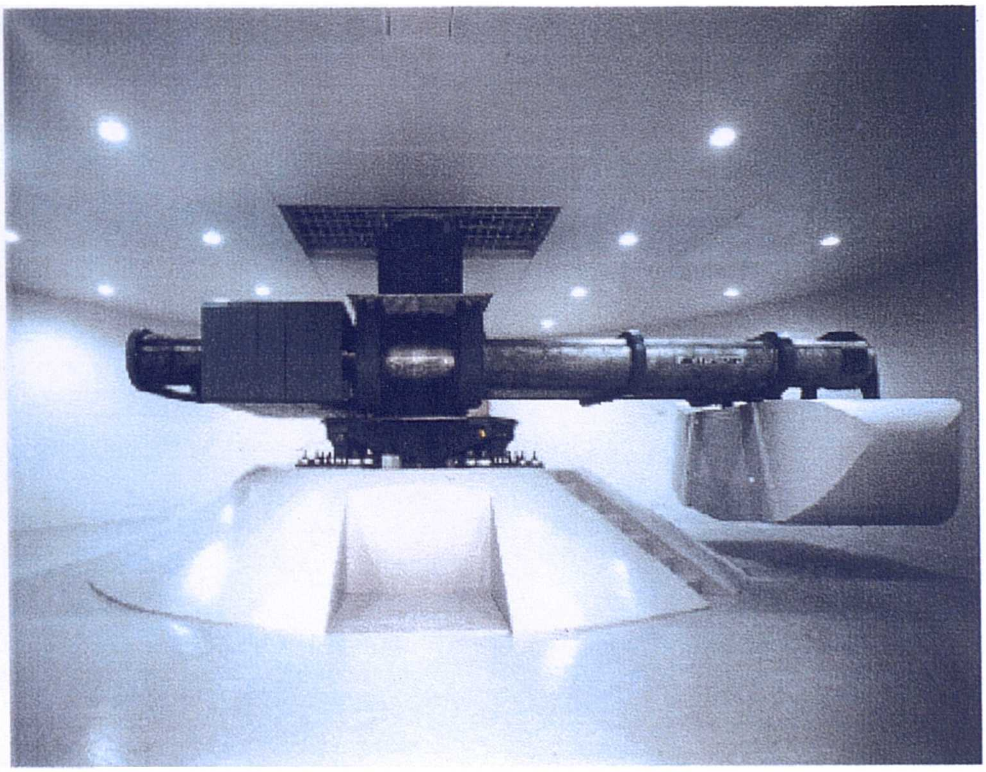


Fig. 3.4 The large geotechnical centrifuge at the Takenaka Corporation, Japan

[From http://www.takenaka.co.jp/r-90/r90e_041.html]

(Original in colour)

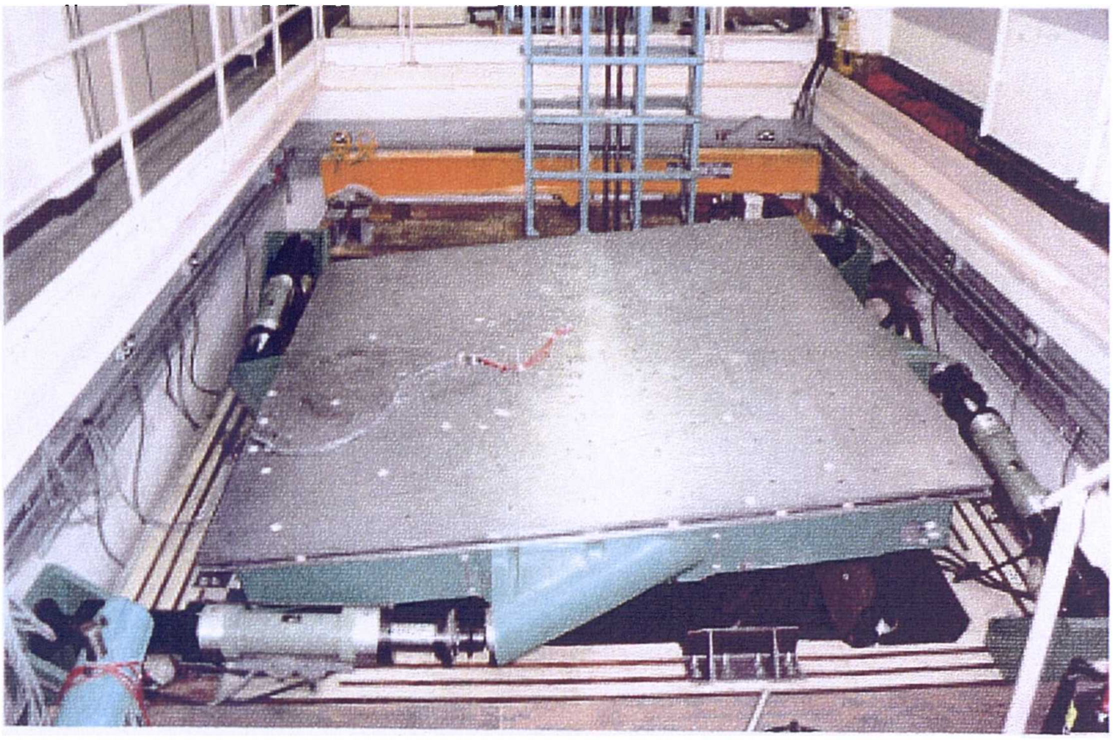


Fig. 3.5 The shaking table at Bristol University

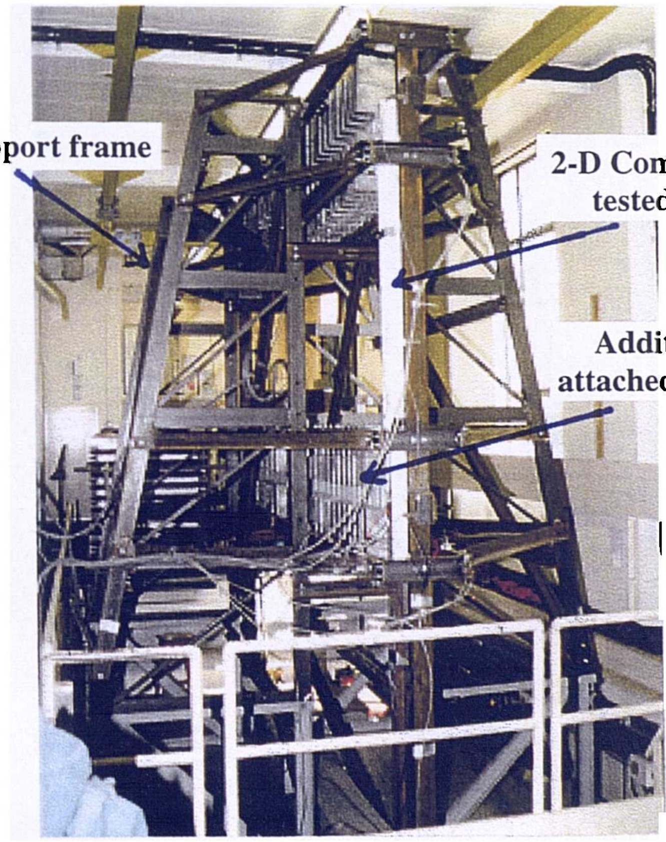


Fig. 3.6 The table at Bristol University testing a composite frame (painted white)

(Original in colour)

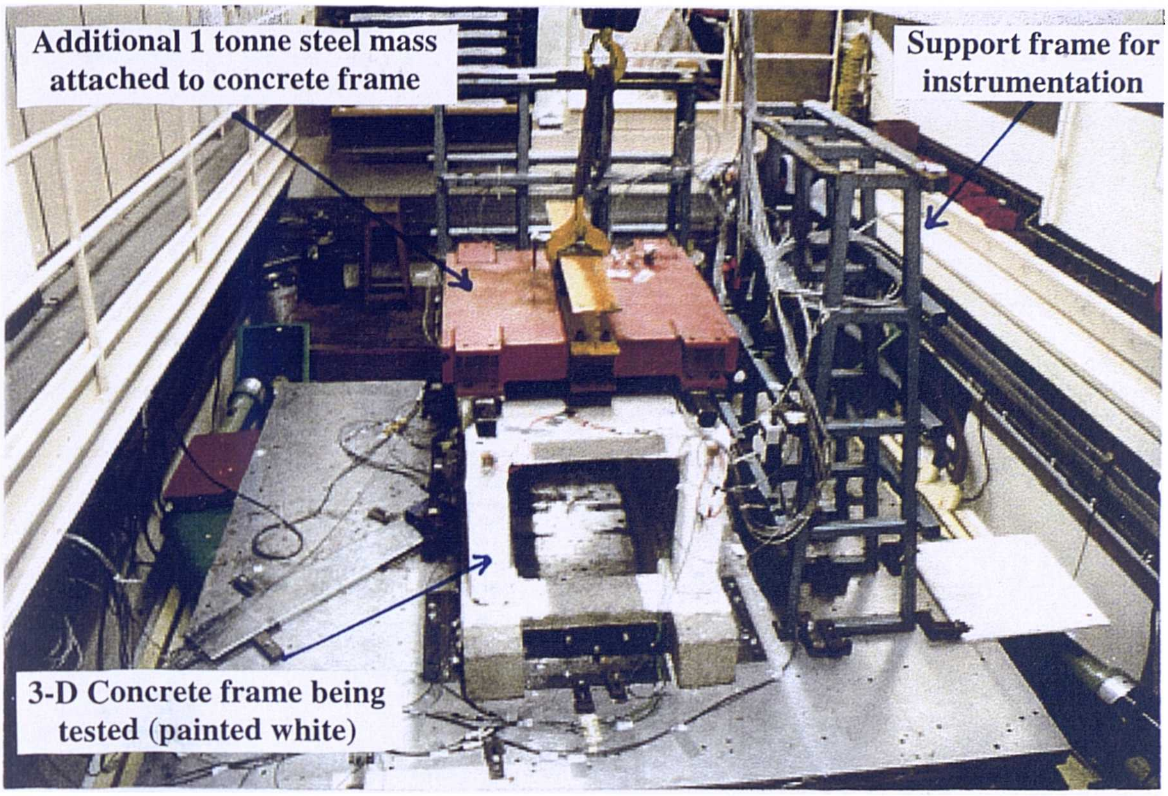


Fig. 3.7 The table at Bristol University testing a concrete frame

Fig. 3.8 The ELSA reaction wall

[From http://www.elsa.jrc.it/elsa/ELSAgeneral.html]

(Original in colour)

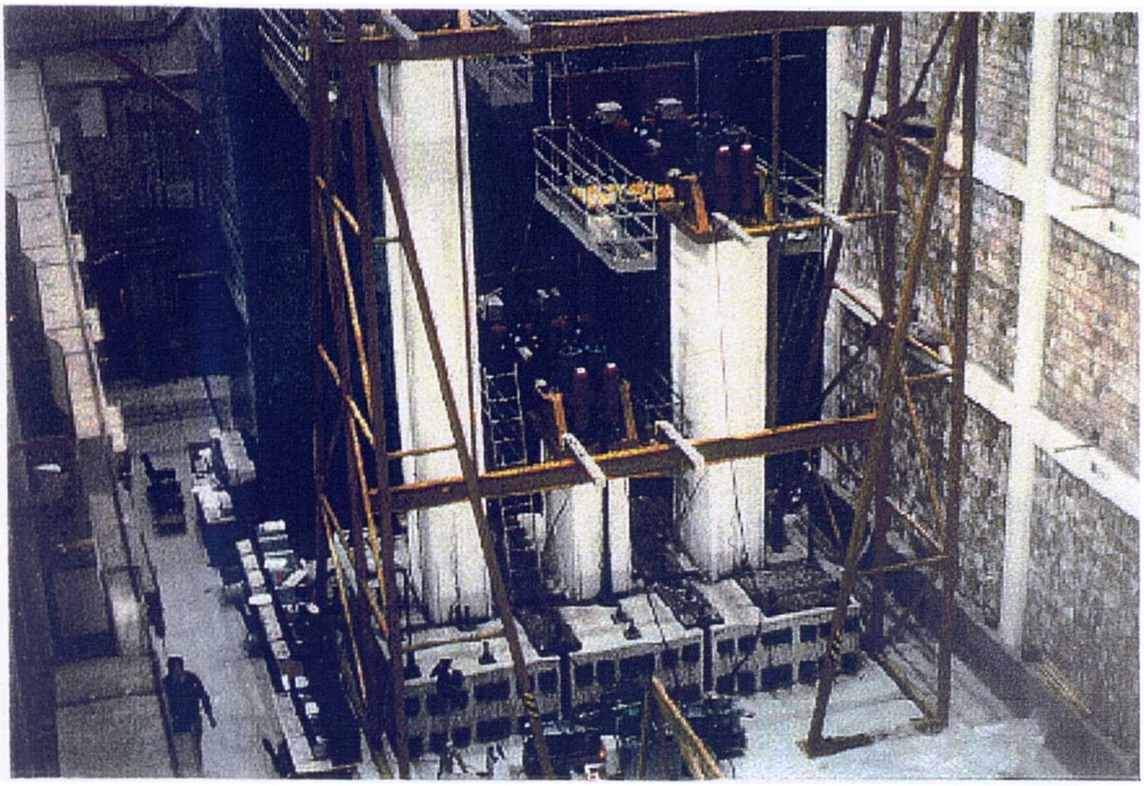


Fig. 3.9 The ELSA reaction wall performing a PSD test of three bridge piers

[From http://www.elsa.jrc.it/elsa/ELSAgeneral.html]

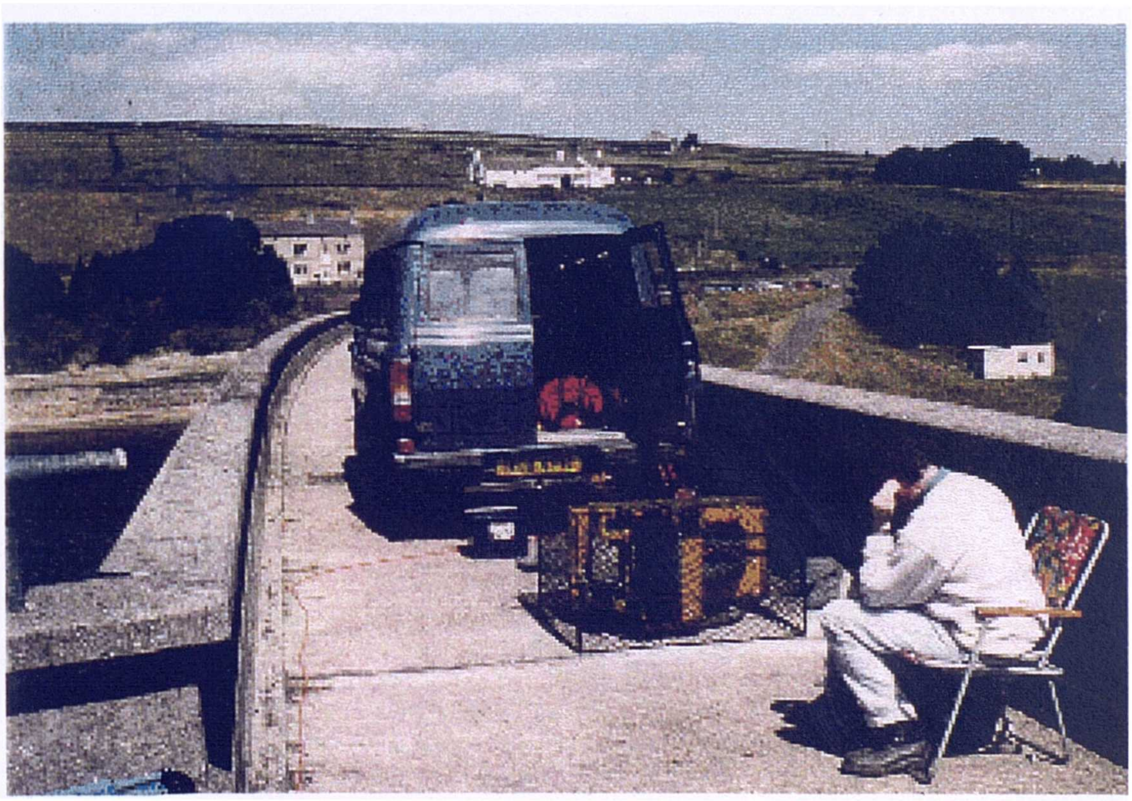


Fig. 3.10 External excitors being used to vibrate a dam

Chapter 4

Shaking Tables as Research Tools

4.1 Introduction

Although shaking tables are only one of the experimental tools available to earthquake engineering researchers they are nevertheless essential because they allow relatively large models to be tested under conditions that are truly representative of real earthquake loading. Once it has been decided that shaking table tests are the most appropriate for a particular research programme, there are two ways of selecting the facility that will be used to perform the tests. In the first case the test may be planned, and then a table with an appropriate capacity and performance chosen to carry out the test. The choice of shaking table can be difficult, and will depend on the specific needs of the research programme. In addition there may be constraints on the access to suitable shaking table facilities. Shaking table tests are expensive, and the larger the facility to be used or the test being planned the more a particular programme is likely to cost. This will often restrict the use of the very large tables to large research programmes where the expense of testing large models can be justified. Also, few shaking table facilities are easily accessible to researchers from outside the group that actually runs and owns the table. A noticeable exception to this was the large European Community (EU) funded "Programme from Human Capital and Mobility - Access to Large Scale Facilities" which funded and actively encouraged participants from throughout Europe to use the larger shaking table facilities that they would not normally have had access to. The second, and more common, option for researchers is that a particular table is available, and that the test therefore needs to be designed around the capacity and performance of that specific table. In either case, researchers need to be aware of the types and characteristics of several or specific shaking tables. This chapter highlights the different types of shaking table that exist, and outlines their basic mechanical characteristics and the common methods of controlling them. Many of the fundamental problems that are associated with all shaking table facilities are also discussed.

4.2 Structural arrangements of shaking tables

There are many possible structural and mechanical arrangements for a shaking table, and all the possible combinations of platform construction and actuator arrangement cannot be covered in detail, but in essence there are four main types of shaking table in common use in the field of earthquake engineering research. These types differ in the number of possible axes in which motion can be controlled, ranging from tables that have movement only in one degree of freedom (DOF) up to tables that can be controlled in all six degrees of freedom (i.e. 3 translational and 3 rotational motions). The most common structural arrangements of shaking tables are discussed below.

4.2.1 Single-axis tables

Single-axis shaking tables are the simplest form of shaking table, where a platform mounted on bearings is shaken by one or more actuators (figure 4.1). These tables are normally orientated to vibrate a specimen horizontally, although some can be adjusted so that vertical motion (only) is possible. For many tests it is only desirable to excite the specimen in one axis as this simplifies subsequent interpretation of the results. In these cases, single-axis tables may provide the best solution for performing the test if they have a large enough dynamic capacity. Single-axis tables are also slightly simpler to control than multi-axis tables. They will still experience most of the mechanical problems outlined in §4.3.2, but the linear bearings commonly used to restrain the motion onto a single axis avoid the problem of coupled horizontal and pitch motions as the bearings are normally very stiff in all but their free axis. The main disadvantage of single-axis tables is their inability to investigate the more complex behaviour of structures under loading in more than one axis.

4.2.2 Two-axis tables

Two-axis tables such as the 20 ft by 20 ft table originally built in Berkeley, California (Rea & Penzien, 1974) are generally designed to allow control over platform movement in one horizontal and one vertical direction (figure 4.2). It can be seen from figure 4.2 that the old actuator arrangement at Berkeley potentially allowed the platform to pitch, and to minimise this pitching, which was not actively controlled by either hardware or software, there are four passive stabilisers which increased the overturning capacity of the table. Another way of controlling the rotational movements is the introduction of an arrangement of torque tubes similar to those incorporated in the LNEC shaking table (§4.2.3).

4.2.3 Three-axis tables with torque tubes

At the next level of sophistication are the three-axis tables which can be controlled in the three linear orthogonal directions. In these tables the rotational components are minimised by the use of three sets of torque tubes (figure 4.3). The great majority of seismic tests employ between one and three axes of linear motion, and by mechanically minimising the rotational components these tables can use fewer larger actuators (a minimum of three) to achieve very good performances at a much reduced cost while still being able to perform the majority of types of test. However, additional mechanical problems, such as flexibility of the torque tubes (§4.3.2.7) and backlash in the extra bearings forming the restraining system (§4.3.2.4), will be introduced into the overall shaking table system reducing the maximum performance of a three-axis table. In addition, the mass of the whole system that has to be excited is larger than in a system with no torque tubes which means that larger electric and hydraulic power plants will be needed to drive the table (§4.3.2.9).

4.2.4 Six-axis tables

The most versatile shaking tables are the six-axis tables like those at the NTU, Athens (figure 4.4) and at Bristol University (figure 4.5) which allow control over all translational and rotational components of the platform motion. However, unlike the restrained three-axis tables, they require a minimum of six actuators, which makes their manufacture expensive, and in addition they require complicated control hardware and software to ensure that the motions in any unwanted axes are held at zero. Although a minimum of six actuators are required to move a six-axis table, eight or more actuators are normally incorporated into the table to balance the static loads, as can be seen in figure 4.4. Because almost all six-axis tables have more actuators than degrees of freedom they are overconstrained. Then, if the actuators are not properly adjusted, calibrated or controlled, there is a tendency for the actuators to start "fighting" and trying to bend or twist the shaking table platform. Apart from potentially damaging the platform this also reduces the effectiveness and maximum performance of the whole system. Three-axis tables with additional restraining systems (§4.2.3), on the other hand, can avoid this problem by only using three actuators, so that there are the same number of actuators as degrees of freedom. However, the additional mechanical problems in the restraining system of a three-axis table will reduce its maximum performance. Since six-axis tables can be actively controlled in all degrees of freedom they have the potential for better performance than three-axis tables. It is also possible with a six-axis table to test structures under every possible form of ground excitation. For example, it is possible to use such tables to investigate the torsional behaviour of irregular plan buildings.

4.3 Mechanical characteristics of shaking tables

In addition to the four main structural types of shaking table that exist there are many different materials and arrangements that can be used for the individual components that form a shaking table. The different types of the main mechanical components of a shaking table are described below, followed by details of the behaviour of each of these components and their effect on the overall dynamic behaviour of a shaking table.

4.3.1 Mechanical components

In this section all the main mechanical components and associated transducers in a shaking table system will be described. Reference should also be made to figure 4.6 where all these components are also shown on a diagram of a typical shaking table. It should be noted that not all these components are used in all tables.

- **Accelerometers** – used to provide information to the control system as to the acceleration of different parts of the platform.
- **Actuators** – servo hydraulic or electro-mechanical jacks that move the shaking table platform. All but the smallest tables use servo hydraulic actuators. Electro dynamic actuators are not normally used in seismic research because of their restricted force capacity. However, small shaking tables using these types of actuators are commonly used by mechanical engineers for fatigue testing of small components.
- **Anchorage locations** – some arrangement of anchorage locations in the surface of the shaking table platform to allow the attachment of specimens. At Bristol a sacrificial top

plate is used to attach the specimens. This 25 mm aluminium plate is drilled and the holes subsequently threaded to accept the necessary bolts.

- **Bearings** – some type of hydraulic or mechanical spherical bearing that allows free rotation of the ends of the actuators.
- **Hydraulic accumulators** – there are several types of oil accumulator used in a shaking table. The smallest are used to smooth the oil flow into and out of the actuator servo-valves. In addition much larger accumulators are often installed on the main hydraulic line that supplies the high pressure oil to the actuators. These accumulators allow short term increased oil flow beyond the capacity of the hydraulic pump.
- **Hydraulic pump** – the pump that provides a high pressure oil flow to run the shaking table.
- **Load cells** – used to provide information to the control system as to the load being applied to the platform by the different actuators. These can be used to stop the many actuators around the platform trying to bend or twist the platform (referred to as "actuator fighting").
- **LVDTs** (Linear Variable Displacement Transducers) – used to provide information to the control system as to the displacement of the different actuators.
- **Nitrogen springs** – these are pneumatic actuators that are connected to large accumulators containing nitrogen at high pressure. These actuators are used to offset the static mass of the platform and specimen, allowing any actuators in the vertical axis to be used solely to provide the dynamic movements of the platform.
- **Preload section in vertical actuators** – at Bristol rather than using nitrogen springs the vertical actuators have an additional low pressure section that is used to offset the dead weight of platform and specimen.
- **Reaction Mass** – provides a very stiff mass against which the actuators can push. Ideally the mass should be 30 - 50 times the mass of the platform + maximum payload. In this way the motion of the reaction mass will be between 2 and 3% of the platform's resultant motion (Clark, 1992).
- **Servo-valves** – valves that control the oil flow into the actuators and hence the movement of the platform.

- **Shaking table platform** – normally a very stiff concrete or steel structure. The shaking table platform at Bristol is slightly unusual in that is made from cast aluminium which makes it much lighter than an equivalent sized steel platform.
- **Shock Absorbers** – devises to damp out any motion generated in the reaction mass during shaking.
- **Suspension system (coil springs or air springs)** – used to lift the reaction block and isolate the shaking table from the surrounding building, they reduce the transmission of high frequency vibrations into the building. Coil springs are more commonly used than air springs to isolate the reaction block from the surrounding building.
- **Test Specimen** – the actual structure or model being tested under seismic loading.

4.3.2 Behaviour of the components of a shaking table

An ideal shaking table is arguably one where the necessary number of axes can be controlled accurately with any required motion and where all the motions in the other axes can be controlled to zero. In addition there should, ideally, be no interaction between the shaking table and the specimen being tested. Unfortunately this is not completely possible, because there will always be some flexibility and non-linearity in the shaking table system which will have to be compensated for by the hardware and software that is used to control the individual actuators in the shaking table system.

The main resonances that are likely to cause difficulties for the control of platform motion are caused by the following flexibilities in a shaking table system (see figure 4.7):

1. The flexibility of the reaction mass on the suspension system / shock absorber system.
2. The internal flexibility in the reaction mass.
3. The local flexibility of support brackets or reaction mass at the connection with the bearings on the actuators.
4. The flexibility and any backlash in the actuator bearings (both at the platform and reaction mass ends of the actuators).
5. The axial and lateral bending stiffness of the hydraulic actuators.
6. The hydraulic oil column bulk modulus stiffness.

7. The axial, torsional and lateral bending stiffnesses of any torsion tubes or other restraining system connected to the platform.
8. The flexibility of the platform itself.

These resonances are particularly important because if the model being tested has a natural frequency close to one of the table resonances it is then possible that the shaking table control system will not be able to compensate for a significant interaction between the behaviour of the table and the specimen. Each of the mechanical resonances highlighted above and a few other issues that effect the performance of a shaking table are discussed in more detail below:

4.3.2.1 Suspension system / shock absorber system

The coil springs or air springs used to support and isolate the reaction mass are, by their nature, very flexible. The resonances produced by the lateral and rocking motions of the reaction mass are likely to be at very low frequencies (≈ 1 Hz to 10 Hz) and within the operating range of a shaking table. For this reason the coil or air springs are normally used in conjunction with a system of heavy duty shock absorbers to create a highly damped resonance. An alternative approach is to keep the reaction mass founded on bedrock and effectively have a reaction mass many thousands of times that of the platform, in which case these resonances do not occur. However, even if the soil and/or bedrock is strong enough, this may not be environmentally acceptable as high frequency noise will be transmitted into the foundations of the building.

4.3.2.2 Internal flexibility of reaction mass

Reaction masses for shaking tables are usually made from heavily reinforced concrete and are generally very stiff. Any natural frequencies are therefore likely to be in excess of 100 Hz and therefore above the operating range of the shaking table. Unless the reaction mass has local weaknesses the flexibility of the reaction mass is unlikely to be a problem.

4.3.2.3 Local flexibility of support brackets / reaction mass at connections

In the majority of shaking tables the bearings at the ends of the actuators are connected directly to the reaction mass (see figure 4.4), in which case if the reaction mass is adequately reinforced there will be little capacity for local flexibility of the reaction mass at the connections. The shaking table at Bristol, on the other hand, has additional supporting brackets between the bearings at the end of the actuators and the reaction mass. These brackets were necessary to allow the table to be installed within the available space. Unfortunately, however, these brackets do introduce another region of flexibility into the shaking table system and if possible this type of arrangement should be avoided, or should at least be made very stiff. The natural frequency of the brackets on the Bristol table is fortunately above 100 Hz, which is beyond the operating range of the table.

4.3.2.4 Flexibility of bearings

Ideally hydraulic bearings should be used in a shaking table as they are axially very stiff while still providing excellent rotational freedom. Unfortunately hydraulic bearings are very expensive, they need a hydraulic oil supply and they also need proper maintenance. This expense can make them difficult to justify although they do eliminate all the problems associated with mechanical bearings. If hydraulic bearings are not used then spherical bearings are commonly used. This type of bearing has a small amount of axial flexibility, and a consequent problem is that if they are not properly adjusted (tight enough to avoid axial movement but not so tight as to restrict rotational movement) then there may be some backlash (opening and closing) of the joints in the bearings when the loads across the bearings change from tensile to compressive forces. Bearing backlash will have the effect of introducing high frequency shock pulses into the platform motion, and as any opening of the bearings is highly non-linear and may not be measured as part of the feedback control loops of the table, it may be impossible to effectively control. A frequent and effective maintenance system should help to minimise problems with the bearings.

4.3.2.5 Axial and lateral bending stiffness of actuators

The actuators in shaking tables are generally quite long and thin because of the requirements for long stroke and high velocity. This means that the actuators will have some axial flexibility and will also have the potential for bending laterally like pin-ended struts. The natural frequency of the axial mode of a normally proportioned actuator is generally high (>100 Hz) while the natural frequency of the bending mode is likely to be lower (generally 2 to 3 times higher than the natural frequency of the oil column §4.3.2.6).

These frequencies are relatively high in relation to the operating frequencies of a table and hence are unlikely to be particularly significant.

4.3.2.6 Oil column resonance

Of all the possible resonances that can be introduced into a shaking table probably the most important is the oil column resonance. This is because it is likely to have a low frequency, often well within the operating range of the shaking table, and is unlikely to be highly damped. The natural frequency of the oil column in any actuator can be calculated from:

$$f = \frac{1}{2\pi}\sqrt{\frac{KA}{Lm}} \qquad \text{(eqn. 4.1)}$$

where

- f is the natural frequency of the oil column (Hz);
- K is the bulk modulus of the hydraulic oil (N/m^2);
- A is the effective cross sectional area of the oil column in the actuator (m^2);
- L is the length of the oil column (m);
- m is the effective mass of platform and specimen being excited by the actuator (kg).

It should be noted that the bulk modulus of the oil is temperature dependant and this should be taken into account when calculating the natural frequency of the oil column. The effect of this temperature dependence can be seen in the table below where the changing bulk modulus causes the natural frequency of an oil column to change by 12% over a range of 30°C.

Table 4.1 Dependency of oil column resonance on oil temperature.

Oil Temperature (°C)	Bulk Modulus (N/m^2)	Natural Frequency (Hz)
10	2.272×10^9	18.720
20	2.110×10^9	18.025
30	1.947×10^9	17.326
40	1.785×10^9	16.590
100	1.315×10^9	14.239

This change in natural frequency of the system may be of particular importance if during a test the oil temperature rises and the natural frequency of the oil columns drops closer to a natural frequency of the specimen being tested. Not only will the dynamic characteristics of the table then have changed but the possibility of table-specimen interaction becomes more likely.

The oil columns in the actuators forming the shaking table at Bristol University have the following properties:

$K = 1.866 \times 10^9$ N/m^2 (bulk modulus of hydraulic oil at operating temp of 35°C)
$A = 3.835 \times 10^{-3}$ m^2 (cross sectional area of the oil column)
$L = 0.42$ m (length of oil column)
$m_{horiz} = 1500$ kg (mass of platform / 2 i.e. 2 actuators in each horizontal direction)
$m_{vert} = 750$ kg (mass of platform / 4 i.e. 4 actuators in vertical direction)
$m_{specimen} = 4000$ kg ((mass of platform + 5000 kg specimen) / 2)

From these figures the natural frequency in the horizontal plane can be calculated as 16.96 Hz which compares with a measured value of 15.00 Hz (see figure 4.8). In the vertical plane, where each actuator excites a smaller effective mass, the frequency can be calculated as 23.99 Hz compared to the measured value of 23.13 Hz (see figure 4.9). As could be expected, the test results show slightly lower natural frequencies than those calculated. This is because the flexibility in the bearings, platform, reaction mass etc. will also contribute to the measured values, lowering the natural frequency of the overall system. When a 5000 kg flexible specimen is attached to the platform the predicted oil column resonance (for a rigid model) is 10.39 Hz while the table shows the actual resonance to be at 12.00 Hz (figure 4.10). In this case the natural frequency of the table is not as low as expected because of the flexibility of the model which is interacting with the table response. In all these three cases the damping value for the oil column resonance is about 5% critical damping. These measurements were made after the DARTEC 9600 digital hardware control had replaced the old analogue controller at Bristol. A comparison of the performance of the two hardware controllers is discussed in §5.5.5.1.

Similar difficulties with oil column resonance were recorded by Díaz & Del Valle (1977) when describing the behaviour of a new shaking table in the National University of

Mexico. They noted the change of frequency with table loading, measuring distortions of up to 15% on their table. However, their test programme did not consider the effect of these resonances on the overall performance of the table. The effect of oil column resonance on the performance of a shaking table was studied as part of this present research, and is discussed further in §5.5.1.1. Methods of controlling this problem are outlined in §6.2.1.

4.3.2.7 Stiffness of any system restraining platform motion

The effect of the axial and lateral stiffness of the actuators in a shaking table on table performance was outlined in §4.3.2.5 above. Similar problems will also occur in any linkages that form any restraining system in a shaking table. For example, the linkages connecting the torque tubes to the LNEC table (figure 4.3) are very long and there is the potential for significant lateral bending here. In the case of the LNEC table, shock absorbing dampers limit the effects of this lateral bending on the performance of the shaking table. An additional problem for those tables that incorporate torque tubes is the torsional stiffness of the tubes themselves. Apart from introducing another resonance into the table system, any flexibility here will place an absolute limit on the extent to which the degrees of freedom that are intended to be restrained actually are. Being a purely mechanical problem that is outside the hardware and software control loops this, as with backlash in the bearings (§4.3.2.5), cannot be compensated for in any way. Good design of the torque tubes and linkages can help to minimise these problems.

4.3.2.8 Flexibility of platform

Apart from the number of actuators around the platform, the different forms of construction of shaking table platforms are probably the most noticeable difference between the many shaking tables around the world. There are two basic varieties, with the platforms of smaller single-axis tables generally being formed from reinforced concrete slabs which are very stiff but also quite heavy. The platforms of larger multi-axis tables, on the other hand, are generally formed from welded steel boxes. There are however some large concrete platforms such as those in the tables at Berkeley, California (Rea and Penzien, 1974) and at State University of New York, Buffalo, USA (Reinhorn and Prawel, 1986). The most important aspects of the table platform are that it is as stiff as possible, and has some method of attaching specimens to it. Concrete platforms are very stiff (Reinhorn & Prawel, 1986), but while the flexibility of the platform does not become an issue the actuators have to be much larger to overcome the inertia of a very heavy platform, which is not normally efficient. Welded steel platforms, on the other hand, will have local plate resonances which may be overcome, to some extent, by having a heavily reinforced structure with many diaphragms and stiffeners. Apart from any local flexibility of the platform it is also important that the platform has a high overall bending and torsional stiffness, and one way of improving on a basic box is to make the platform in the form of an inverted pyramid, as are the Bristol and LNEC shaking tables (figures 4.5 and 4.3). For a well designed table the first natural frequency of the platform itself is well above the normal operating range of the table.

4.3.2.9 Mass of platform

The last main consideration is the mass of the platform, and here there are two conflicting issues. In the first place a very light platform will require smaller actuators to move it, which reduces the initial cost of the facility. However, a very light platform is much more likely to be affected by significant table-specimen interaction (§5.4.4) which will require a better hardware and software system to effectively control. Therefore the larger the platform mass the better it is from the point of view of the specimen, but this requires larger actuators, increases running costs and can limit the maximum accelerations achievable. One solution to this conflict is the use of a light platform with high capacity actuators. For small light specimens the platform is used as is, but when a larger specimen that might interact significantly with the table is to be tested then additional static mass, up to the capacity of the table, may be added to the platform. This additional mass will then help to reduce the table-specimen interaction by increasing the platform mass that has to be excited by the specimen. The improvements in table performance that can be made by the addition of static mass to a platform are shown in more detail in §5.5.1.1.

The other effect of having a larger, more massive platform is simply the increasing capacity of the shaking table to deal with larger scale models. In §3.2.1 part 4 it was shown that the larger the model the simpler the scaling issues become. Therefore, if the cost of construction of the shaking table, the running costs and the cost of manufacture of models is not a problem, then the larger the mass of the platform and capacity of the table the better.

4.3.2.10 Other mechanical and electrical problems

In addition to the resonances discussed above there will also be some non-linearity in the behaviour of the servovalves (Kusner et al., 1992) and the instrumentation (accelerometers, LVDTs) that are being used to monitor the motion of the platform and to close the basic feedback control loops (MTS, 1985).

4.3.3 Conclusions

Unfortunately very few researchers will have the opportunity to try to minimise or eliminate some of these problems when designing a new shaking table or working on the upgrade of an existing one. Even so, all the difficulties mentioned above would still be present to some extent in even in a new shaking table. The most important thing for researchers is that they are aware of the mechanical problems that are inherently present in any shaking table system and ensure that these issues are taken into account when any experiment is being planned.

4.4 Controlling the motion of shaking tables

Initially it might seem that single-axis tables, being mechanically the simplest, would be straightforward to control. However, such tables still have all the problems associated with servo-hydraulic actuators, bearings etc. and it has been shown by Takahashi et al. (1974) that even controlling what might appear to be a simple system is still very difficult. However, before considering the methods that can be used to control shaking tables it is worth looking at the types of motion that are normally used in seismic testing.

4.4.1 Types of motion normally used

There are many different types of shaking table motion that are normally used for research purposes. These are:

Sine waves and sine dwells – These can be most effective in the study of very complicated structural systems where it is desirable to keep the response of the structure being tested as simple as possible. Figure 4.11 shows a typical input motion that has been used at Bristol to study the behaviour of simple foundations on a soil deposit. By keeping the excitation very simple it becomes much easier to compare the results of any experimental work with the theoretical studies for these type of systems. With this type of motion it is also possible to transfer all the energy of the shaking table directly into the structure at its natural frequency, and this can produce a much more severe test for the specimen. In this way even a small capacity table can test a specimen very severely. However, because this type of motion does not have a broad frequency range, it will normally only excite one mode of vibration in the structure which, while being very helpful in understanding the basic behaviour of the structure, may be misleading when considering the behaviour of the structure under actual earthquake loading.

Impulse tests – These types of input (figure 4.12) can be used to characterise the dynamic behaviour (natural frequencies and damping ratios) of a structure on a shaking table. However, they are rarely used in shaking table testing as they can create very high stresses in the table itself as the actuators are run at their maximum force capacity. The use of a random noise signal (see later) will produce an equally good, if not better, structural characterisation and this type of motion is less potentially damaging to a shaking table.

Sine Sweep – This type of input (figure 4.13) is generally used to determine the natural frequencies and damping values of a test structure. However, structures with low damping can easily be damaged when the signal frequency reaches one of the natural frequencies of the model. Performing a test with this type of signal can also be quite time consuming, so the use of a random noise signal to characterise the model is preferred.

Random noise – This type of input is generated by combining a broad band of frequencies with random phases into a single signal. A typical section of drive time history generated in this way is shown in figure 4.14. This sort of input motion can be very useful in determining the natural frequencies and damping of the structure being tested. Since the structure is not being continually excited at a natural frequency it is less likely to be over excited and damaged by the test. Also, because all the natural frequencies of the model are being excited, it is much quicker to use this type of input compared to sine sweep inputs to characterise a test model.

Actual earthquake records – Over the last 50 years a great number of real earthquakes accelerograms have been recorded, for example the classical 1940 N-S El Centro (California) record (see figure 4.15) that has been used in a great deal of research over the years. These generally consist of three orthogonal acceleration records. These records may come from sites with completely different ground conditions (including hard rock or soft soil sites) and while these are real earthquakes the experimentalist must be aware that they may not contain the appropriate frequency content for the test being planned, so the frequencies may require scaling. Another problem with these records is that the signals may be clipped (particularly if the instruments were close to the epicentre of the earthquake). The records are also likely to be unreliable at low frequencies as the accelerometers used to record the data are generally not able to record any static or very low frequency displacements in the ground. Even with all these problems, real earthquake time histories are often used in shaking table tests as they may be felt to be more realistic than the artificial earthquakes that are used for designing structures.

Artificial earthquakes – The design spectra in most contemporary earthquake codes (UBC, 1985 & SEAOC, 1985) are designed to envelope many real earthquakes and are then smoothed to produce simple spectra with only a few defining points. A typical design spectrum that has been used for a shaking table test is shown in figure 4.16. These spectra are normally defined in terms of ground accelerations, and the acceleration time history generated to match such a spectrum must be converted into a required platform displacement by double integration.

Summary – The choice of input motion(s) for a particular shaking table test will depend very much on the desired outcomes of the test, for example is accurate reproduction of the acceleration or displacement response of the platform most important? The issues surrounding this choice are covered more in §6.6.

4.4.2 Standard control techniques

All shaking tables incorporate two distinct types of system that are used to control the platform movement. These systems are often referred to in several different ways but for simplicity they will subsequently be called the 'Hardware' or 'Inner loop' control system and the 'Software' or 'Outer loop' control system.

Three of the four shaking tables studied for this research have actuator arrangements similar, or fairly similar, to that shown in figure 4.4. This arrangement of eight actuators creates three main problems from the point of view of controlling the motion of the shaking table platform. Firstly, there are more actuators than degrees of freedom so some technique must be used to stop the actuators fighting each other (§4.2.4). Secondly, a control system is needed to convert the desired platform motion into the eight different signals needed to drive the actuators. Finally, although the arrangement of actuators shown in figure 4.4 allows movement in all six degrees of freedom, the majority of shaking table tests require motion in a single axis only, although two-axis and three-axis tests are also quite common. Tests that require motion in fewer than six axes present some problems, as movement must therefore be constrained in one or more axes. This restriction on platform movement and specific control over the motion of each of the actuators is generally provided for within the hardware control system. However, this research has shown that current hardware control systems cannot always restrict motion in the unused axes to acceptable limits. In this case an additional software control system must be used to compensate for any inadequacies in the hardware.

4.4.2.1 Analogue / digital hardware control system

This is the electronic hardware that controls the individual actuators in the shaking table and incorporates the feedback loops such as those shown in figure 4.17. These feedback loops are basically enhanced and extended versions of a simple proportional feedback loop (figure 4.18). Simple feedback loops like this work by subtracting the actual position of the actuator from the desired position and then using this new signal to drive the servo valve on the actuator. This simple case is called "proportional feedback" because the magnitude of the signal that controls the servo value is proportional to the difference between where the actuator should be and where it actually is. The speed with which the actuator responds can then be varied by adjusting the "gain" or "amplification" that is applied to the control signal. The more complicated feedback loops in some shaking table systems use additional feedback terms, each with its own gain, to improve the speed and accuracy of the actuator response across a wide range of frequencies. The process of adjusting the various gains to optimise the response of an actuator is called "tuning". In addition to the feedback loops that control individual actuators, the hardware controlling the whole table may also incorporate additional feedback loops that feed, for example, some of the horizontal platform motion into the pitch axis to try compensate for one form of table-specimen interaction. The tuning process then allows the whole table system to be adjusted to minimise table-specimen interaction. Normally the hardware control system is "tuned" with the specimen on the platform, but at low vibration levels because the tuning must take place before the specimen is tested, yet must not damage the specimen. The ideally tuned servo-controlled system would have unit gain and no phase difference between the drive signal and the platform response in each axis, across the full range of operating frequencies, both with and without a specimen on the platform (i.e. the platform response exactly follows the desired input signal). It is important to realise that even if the hardware control system could be tuned so that the frequency response of the shaking table platform was unity at all frequencies in all excited axes and zero in all other axes, an additional software control system would still be required to compensate for any non-linear response of the specimen.

An example of the actual performance of different types of these feedback loops can be seen in a comparison between the responses of the very basic analogue hardware that controlled the Bristol shaking table in 1993 when this research started (figure 4.19) and the hardware controlling the Athens table (figure 4.17). The Bristol hardware only incorporated two feedback loops, displacement and acceleration, for each actuator. Each of these two feedback loops incorporated one user adjustable potentiometer that could be used to adjust the amount of each of these two signals that formed the overall feedback signal. A typical system transfer function (should ideally be 1.0 across the entire frequency range) for the Bristol table before any tuning is shown in figure 4.20, and after tuning in figure 4.21 (details of the abbreviations on the figures can be found in Appendix B.2). It can be seen that although the performance of the system has improved (the amplitude of the resonance has reduced), the response of the shaking table is still far from ideal. Similar results from the Athens shaking table before and after tuning are shown in figures 4.22 and 4.23. The Athens table incorporates the more comprehensive MTS feedback system (Crewe et al., 1996), and comparison between the results obtained in Bristol and Athens show that the MTS feedback system is much more effective at minimising system resonances. However, the system transfer function at Athens is still not completely flat, and to compensate for these errors between the actual and ideal platform response an additional software control system is used.

The hardware controlling the shaking table may also allow different types of input signal to be used to actively drive the shaking table. The two standard types of input motion used to drive shaking tables are displacement signals and acceleration signals. In the case of "displacement control" the arrangement of feedback loops is adjusted so that the primary feedback loop is the platform displacement. In the case of "acceleration control" the primary feedback becomes the platform acceleration. Generally if accelerations are actively controlled they are reproduced on the platform much more accurately than the desired displacements, and vice versa, if displacements are actively controlled the acceleration response of the platform will not be so accurate.

4.4.2.2 Software control system

This is the software that is used to compensate for any inaccuracy in the tuning of the hardware control system. In theory, this software allows the desired motion to be accurately reproduced on the shaking table platform. Current software for controlling shaking tables does this by an iterative process of recording the platform motion achieved during a seismic test and then correcting the drive signal so that the platform motion for the next test will be closer to the required motion. This procedure will compensate for any errors in the tuning of the hardware and can also compensate for any linear shaking table-specimen interaction, but being an iterative procedure that does not react in real-time, it cannot compensate for any non-linear response in the table or specimen that occurs during a particular shake. There are, however, some new developments that are currently being made in the creation of real-time control software that can automatically compensate for all of these time dependant changes as they actually happen. More details about these new control techniques are given in §6.7 and some possibilities for the future of shaking table testing are outlined in §6.8.

A schematic for a whole shaking table control system is shown in figure 4.24. The hardware control system is completely passive once it has been tuned. The software control system then has three functions. Firstly (A), it sends a drive signal to the shaking table via the hardware control system and secondly (B) it records the platform motion. Once the shake is completed the software is used to compute a new drive signal (C) that should make the platform move as required. This iterative process of sending a drive signal then modifying it for the next shake should eventually cause the platform to behave as required. An example of how effective this software iteration can be is shown in figures 4.25 and 4.26. Both figures present five graphs showing a single-axis response from the shaking table at Bristol. The top two graphs show the required and achieved platform time histories; in this case these are accelerations. Beneath these two graphs is a third graph in which is an overplot of the two time histories which highlights any differences in the signals. Finally, the lowest two graphs show the amplitude and phase, respectively, of the frequency response function between the achieved and required time histories. Ideally, the amplitude plot should be unity and the phase should be zero in the significant frequency ranges if the desired and achieved platform motions are identical. Figure 4.25 shows a first attempt at reproducing the El Centro earthquake on the Bristol table. This first test does not give a very accurate reproduction of the time history but after the software has been used to modify the drive signal three times then a much better platform motion has been achieved (figure 4.26). The El Centro earthquake record shown here has over 95% of its energy in the frequency range from about 1 Hz to 8 Hz. Above 8 Hz, the record contains very little energy and therefore the frequency response function is subject to spurious peaks arising from numerical noise effects beyond this range; this is discussed in more detail in §5.4.7.

Details of the methods commonly used in Bristol for calculating the new drive signal at each iteration are shown in figures 4.27, 4.28a and 4.28b. These methods work in the frequency domain and rely on the Fast Fourier Transform (FFT) and inverse FFT to convert the time history into the frequency domain and the modified amplitudes and phases at the varying frequencies back into the time domain. The time histories are segmented into overlapping blocks, the inverse transfer function for each block computed and an updated drive signal for each block is created. This approach leads to a time dependant transfer function that can deal with repeatable non-linearities in the shaking table system itself, but at the expense of the frequency resolution of the transfer function.

The iterative time history matching methods used on other shaking tables work on the same principle but tend to vary slightly in detail. For example, in some systems a measurement of the transfer function of the shaking table is made at low to moderate amplitudes for the particular test configuration and the driving signals are then pre-compensated using the inverse transfer function. Broad band random noise is applied to the shaking table for several minutes, preferably with the specimen mounted, and an averaged inverse transfer function is computed. This averaged transfer function is used to generate an initial estimate of the required drive signal. Once the actual platform motion has been recorded an inverse transfer function for the error between the acquired and required time history is computed and an updated drive signal created.

The software control systems for all shaking tables can leave the control of axes requiring zero motion to the hardware controller or can apply drive signals to those axes to force no movement. The software control of axes requiring no movement then proceeds in the same way as the axes *being matched to specific motions except that the required motion in those* axes is now zero throughout the shake.

All these software control methodologies rely on the fact that the shaking table and specimen being tested form a linear system. Consequently, the current control systems may not accurately reproduce the required platform motions under some load conditions. Most dynamics research using test specimens investigates structural behaviour well beyond the elastic range of the materials, and when the material properties change as they enter their plastic region, the overall dynamic characteristics of the specimen also change. This can have a knock-on effect on the response of the shaking table when the specimen mass is similar to or greater than that of the table platform.

4.5 Appropriate use of shaking tables

At first sight it might appear that using six-axis tables will always be the preferred option when performing seismic testing as they provide the most comprehensive type of test facility. However, for research purposes it is generally undesirable to perform testing of a specimen with defined motions in all six degrees of freedom. This is mainly because the resulting model behaviour from such a test would be very difficult to understand fully and to compare with analytical results or data from field tests. Therefore the majority of tests are single or two-axis tests with the motion in the other degrees of freedom being held, ideally, at zero. Tests that require motion in fewer than six axes then present some problems as movement must be constrained in one or more axes. This restriction on platform movement can be performed in two ways. The first possibility is to use a shaking table which is mechanically restrained to move in only the desired directions. In these tables the potential problems of controlling all the other degrees of freedom to zero do not occur, assuming that these motions are adequately restrained by systems of bearings or torque tubes. However, it is impossible to control any movement that does actually occur because the restraining system is not completely rigid, and these uncontrollable motions may have an undesirable effect on the test results. The second option is to use a shaking table that is able to move in more axes than are required for a specific test. In this case various techniques to limit the unwanted axis motions are used within the control hardware. However, this research has shown that current hardware control systems cannot always restrict motion in the unused axes to acceptable limits. In this case the control software must be used to compensate for any inadequacies in the hardware.

Neither of these two options completely solves the problem of accurately controlling the motion of a shaking table platform during a seismic test. In one case passive restraint systems are used, and in the other active control systems, but neither system is completely foolproof, so all shaking tables tests therefore become a compromise between ease of use and accuracy of control. However, for some single-axis tests, there may be significant advantages in using a comparatively simple single-axis table as compared to the more complex six-axis systems.

4.6 Conclusions

There are many different types and size of shaking table currently in use throughout the world, and these shaking tables are used to perform much of the dynamic testing for earthquake engineering research. With so many tests now being performed on different shaking tables, it is important that researchers are using facilities suitable for their particular tests. The importance of selecting an appropriate shaking table facility was briefly mentioned, in general terms, in the introduction to this chapter. From the purely research viewpoint it is preferable to select a facility which has the precise capabilities required for the particular experiment. For example, if a single-axis test is required then it may be unnecessary to seek a facility that has a full six-axis table. On the other hand, it is also important to consider the dynamic capabilities of the table, the instrumentation and acquisition systems and any other facilities available, and these are usually better at the larger establishments. Finally, in addition to selecting the most suitable shaking table facility, the researcher must be confident that the table is performing effectively and also to a common standard, so that the results obtained will be directly comparable with those of other workers. The current growth of, for example, pan-European research groups working together on very large research programmes and using many different facilities, emphasises the need for such confidence. The performance tests described and discussed in the following chapter were designed to assist with overcoming these problems.

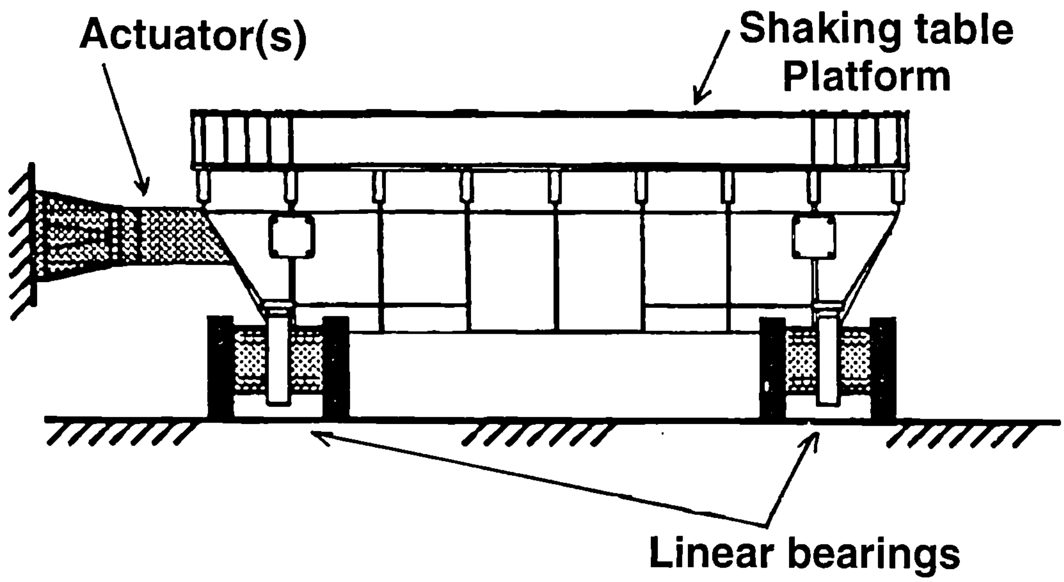


Fig. 4.1 Typical arrangement of a single axis table

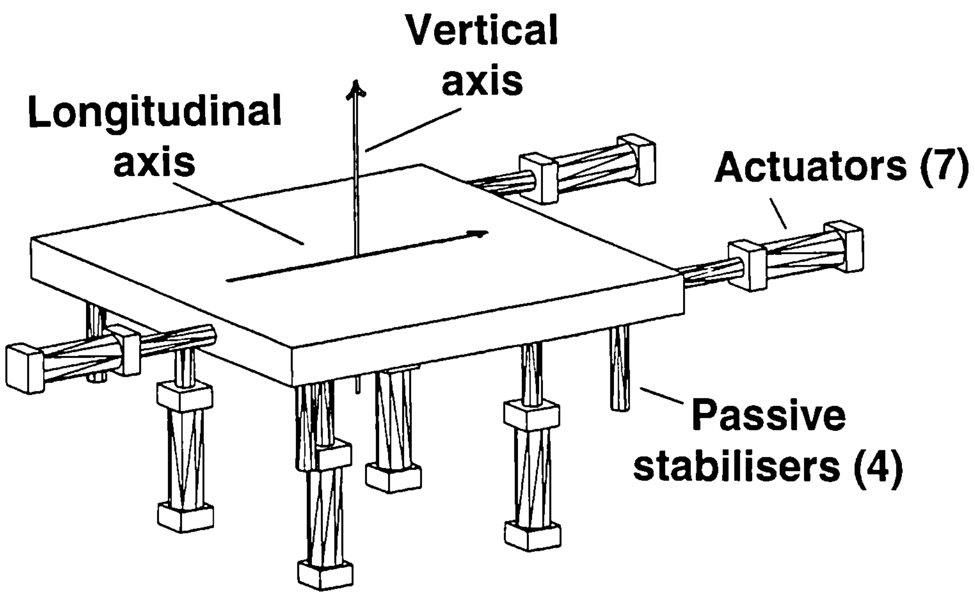


Fig. 4.2 Typical arrangement of a two-axis table

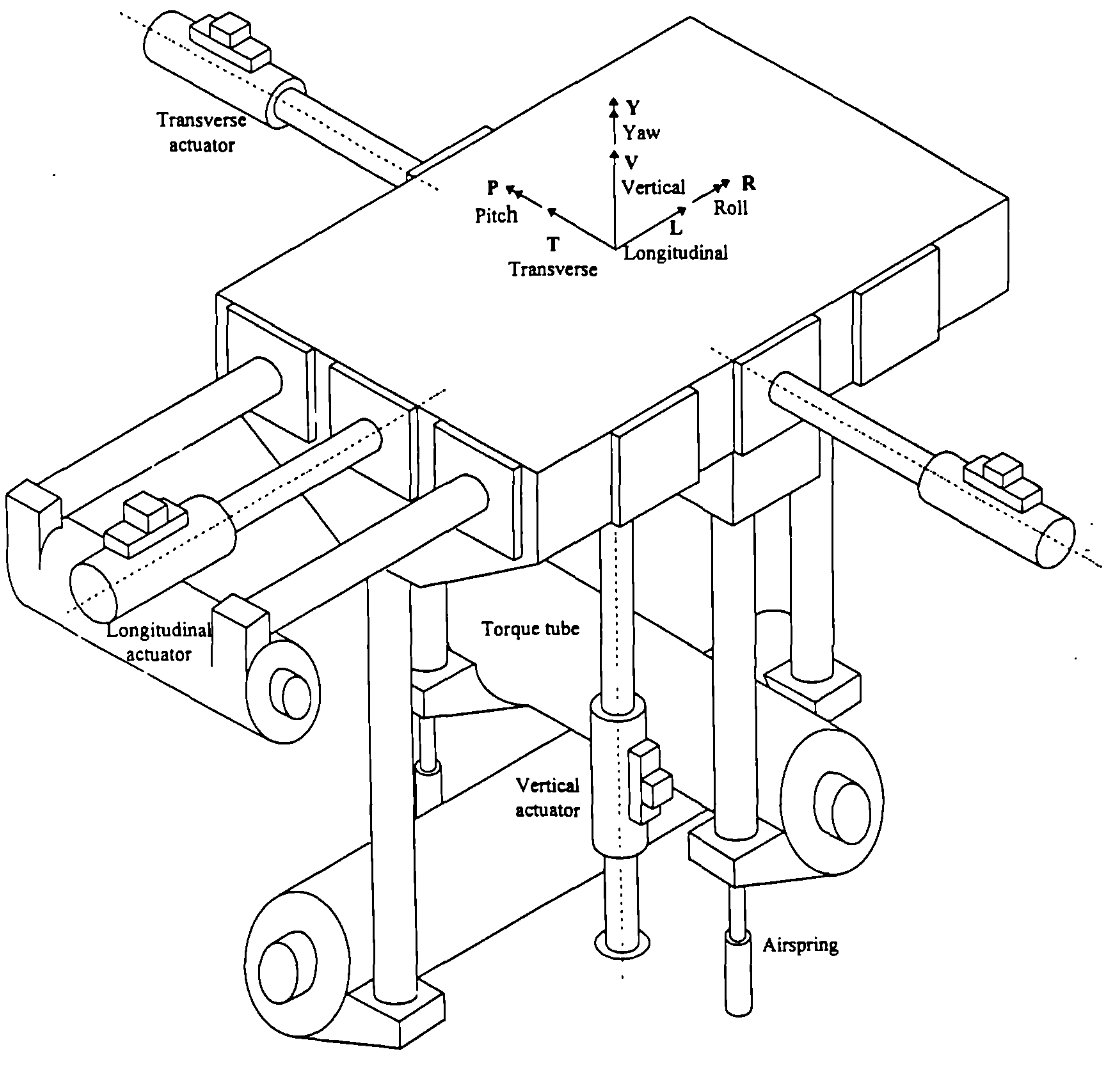


Fig. 4.3 Arrangement of the three-axis table at LNEC

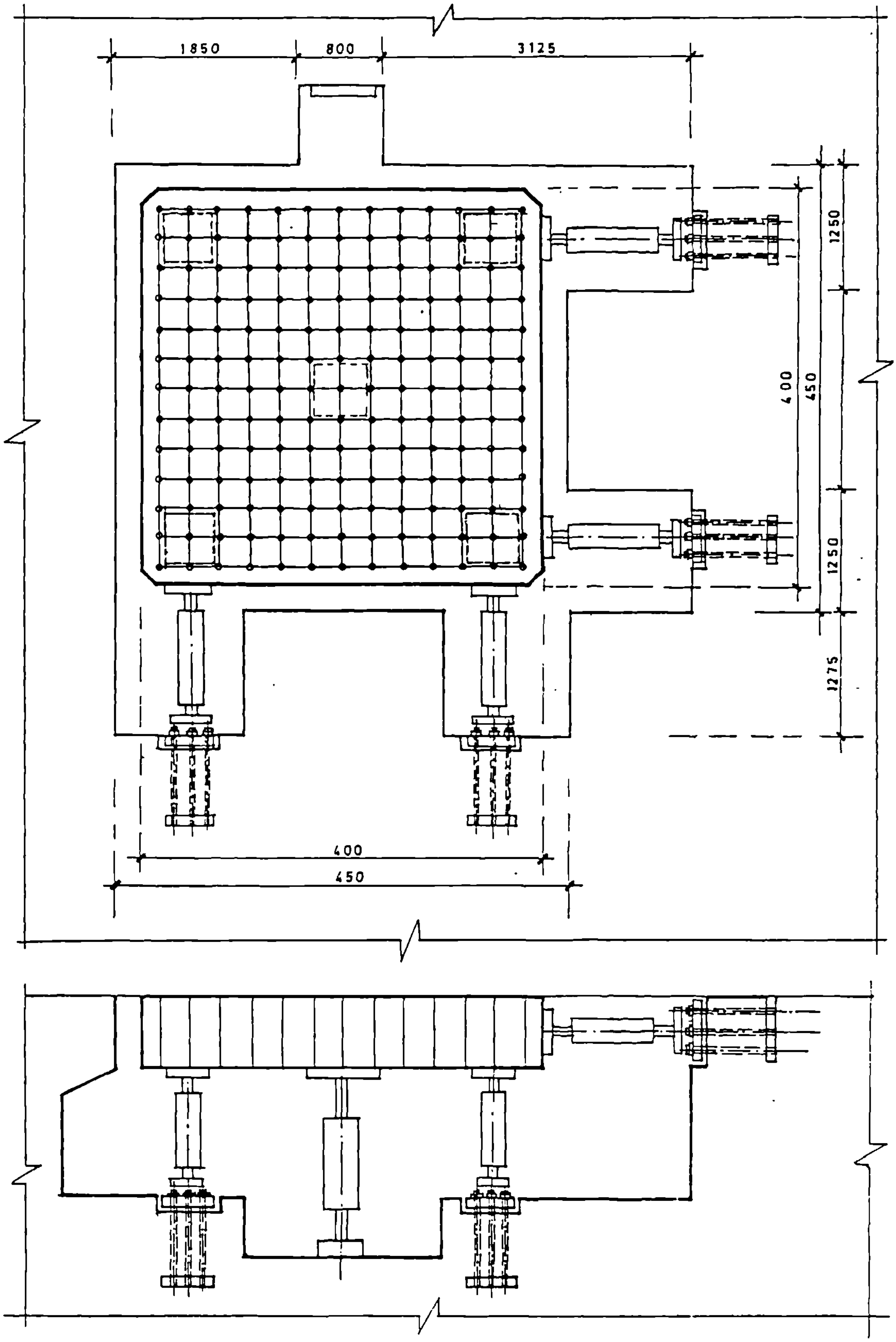


Fig. 4.4 Arrangement of the six-axis table in Athens

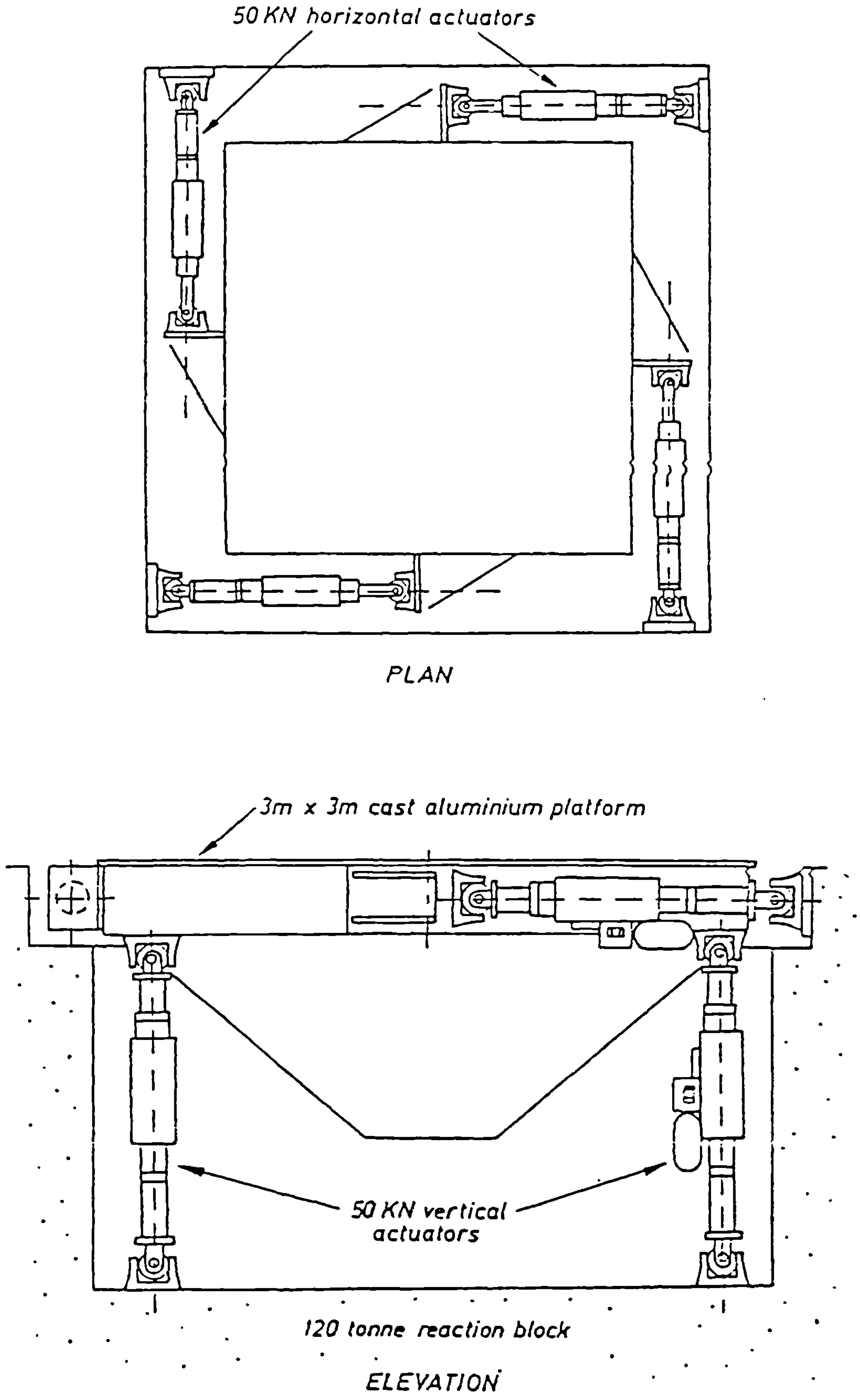


Fig. 4.5 Arrangement of the six-axis table in Bristol

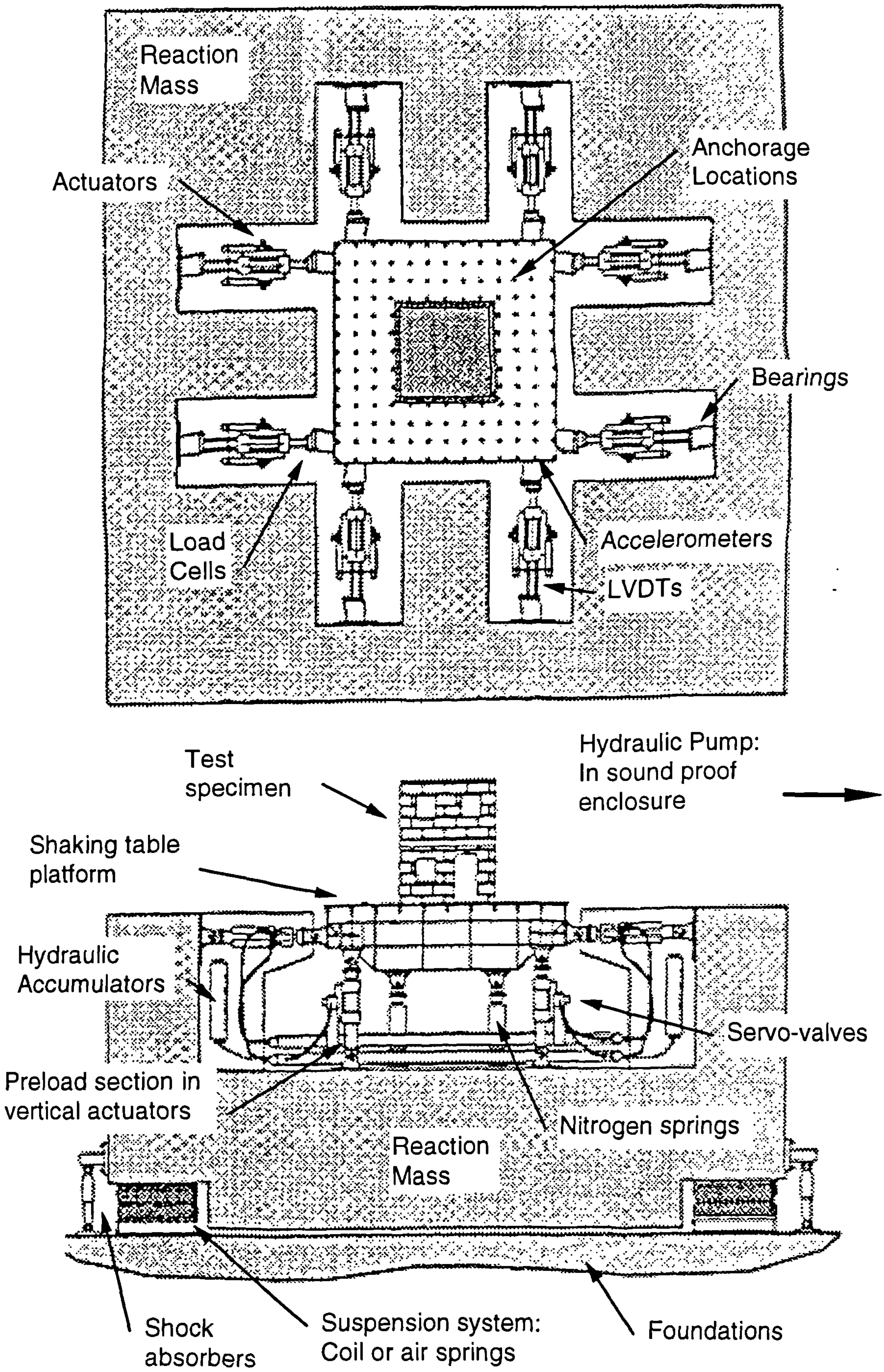


Fig. 4.6 Components of a typical shaking table

1 The flexibility of the reaction mass on the suspension system / shock absorber system.

2 Internal flexibility in the reaction mass

3 Local flexibility of support brackets or reaction mass

4 Flexibility and any backlash in the actuator bearings

5 Axial and lateral bending stiffness of the actuators

6 Hydraulic oil column bulk modulus stiffness

7 Axial, torsional and lateral bending stiffnesses of any torsion tubes or other restraining system

8 Flexibility of the platform

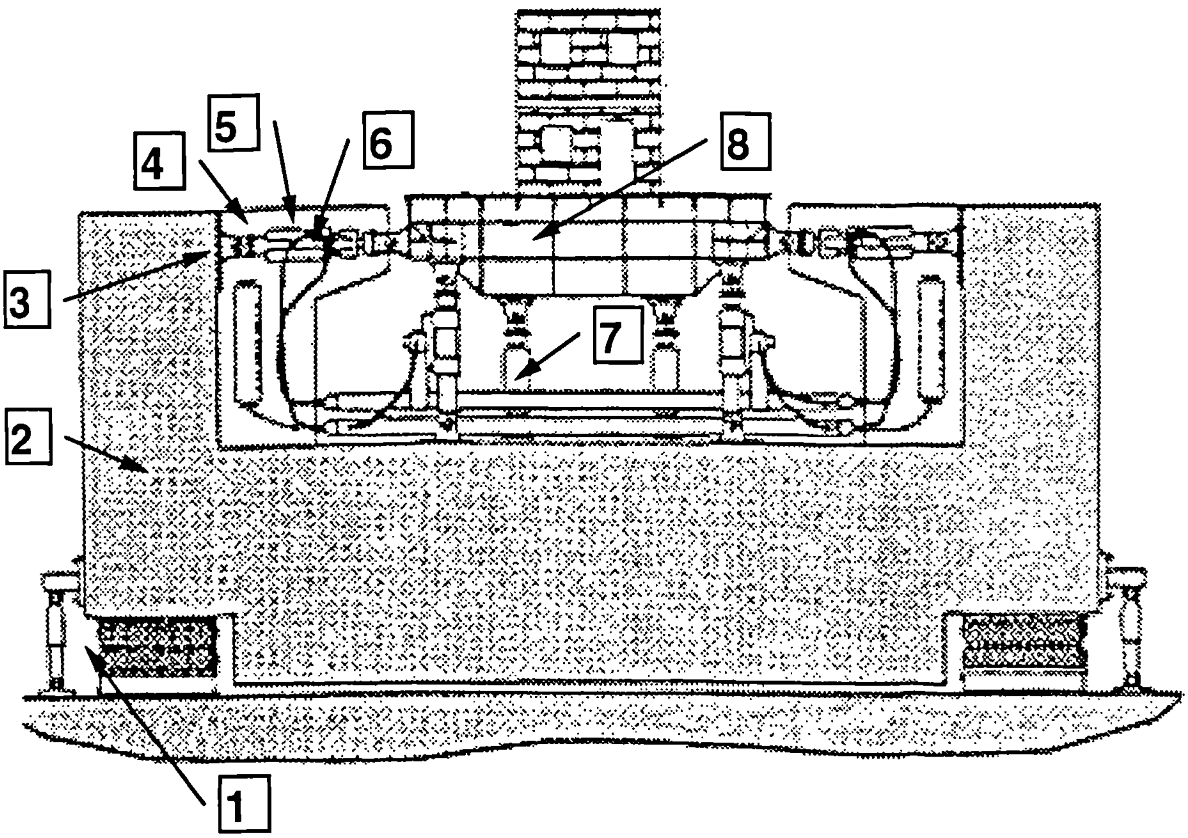


Fig. 4.7 Potential regions of flexibility in a typical shaking table

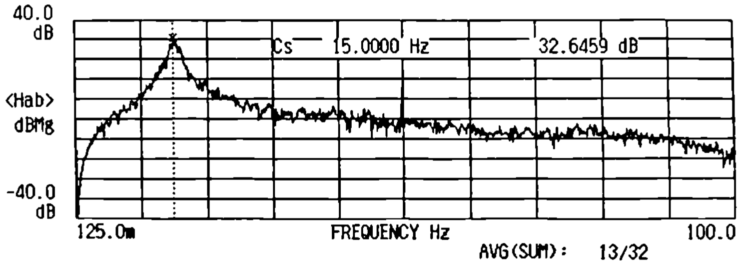


Fig. 4.8 Oil column resonance in the horizontal axis of the Bristol shaking table

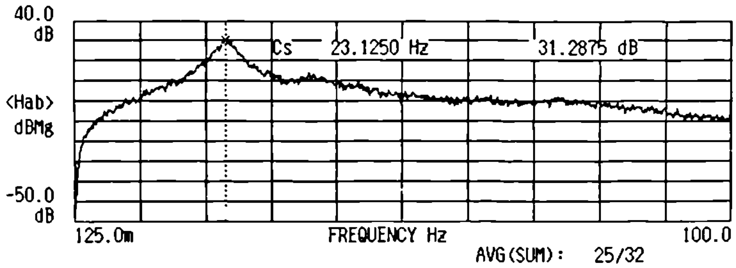


Fig. 4.9 Oil column resonance in the vertical axis of the Bristol shaking table

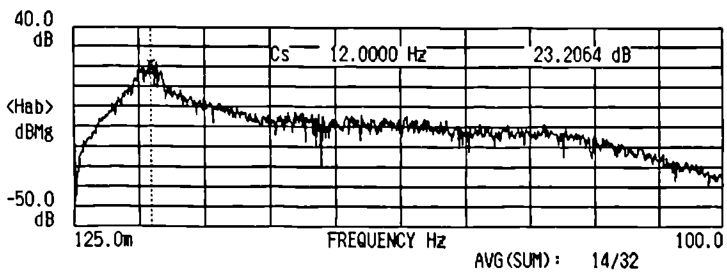


Fig. 4.10 Oil column resonance in the horizontal axis of the Bristol shaking table with the 5 tonne specimen attached

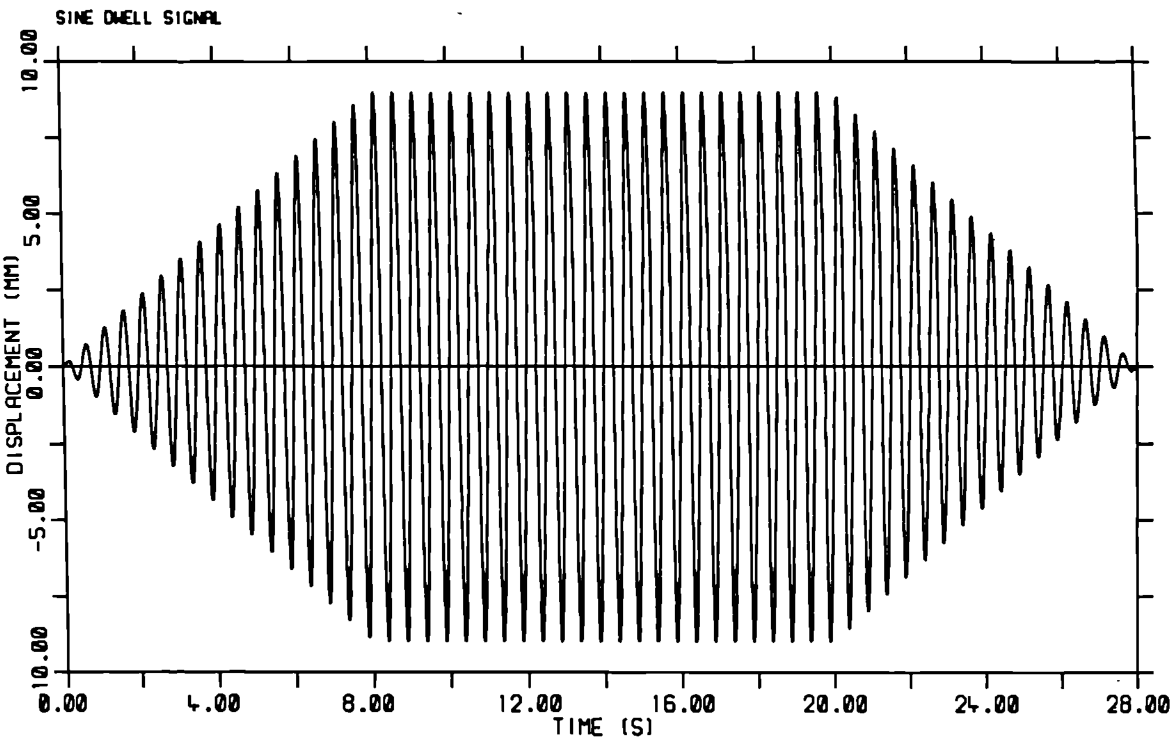


Fig. 4.11 Typical sine dwell signal used for soils tests at Bristol

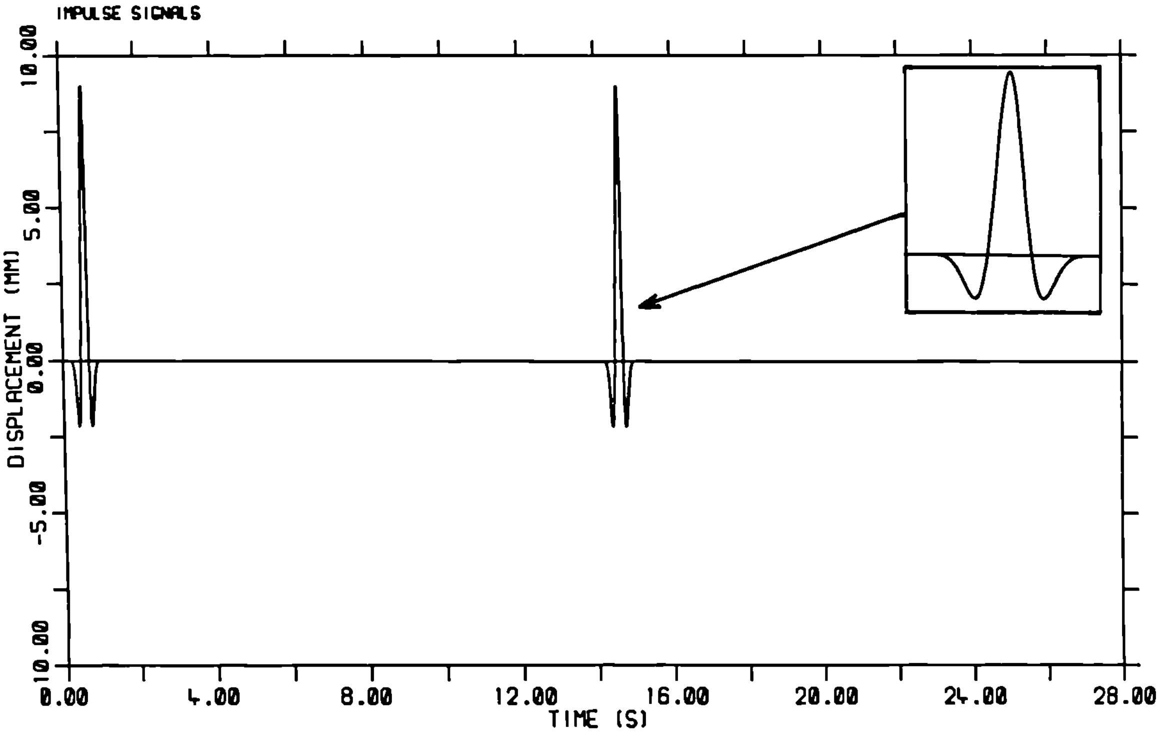


Fig. 4.12 Typical impulse signal

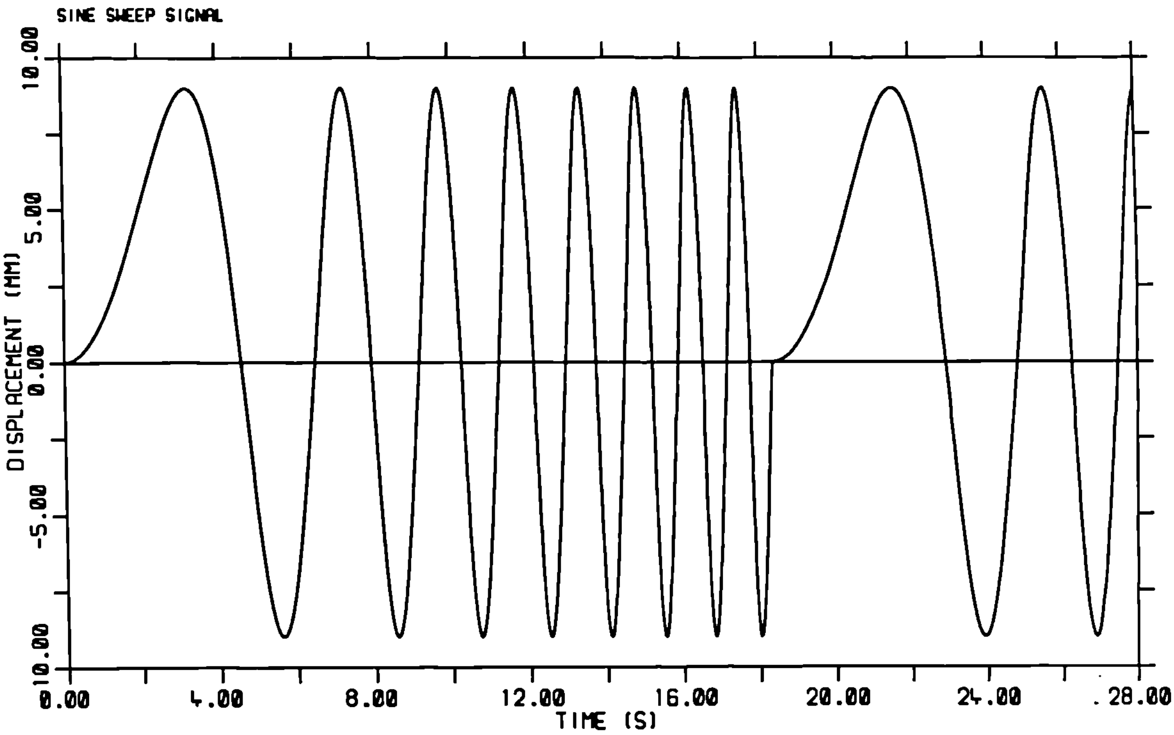


Fig. 4.13 Typical sine sweep signal

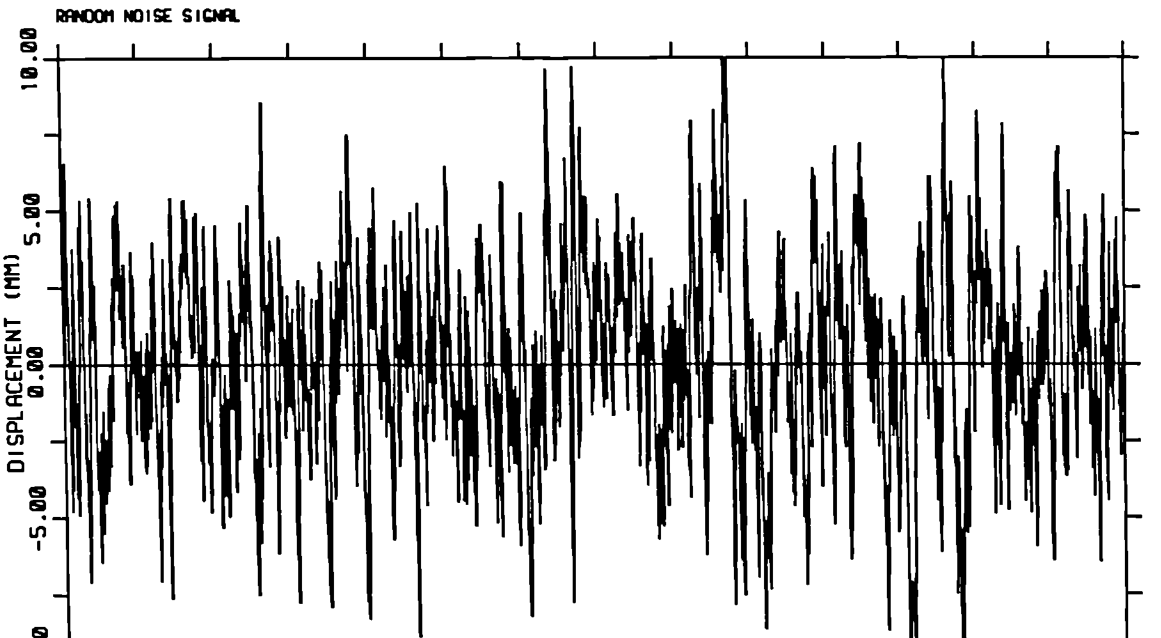


Fig. 4.14 Typical section of a random noise time history

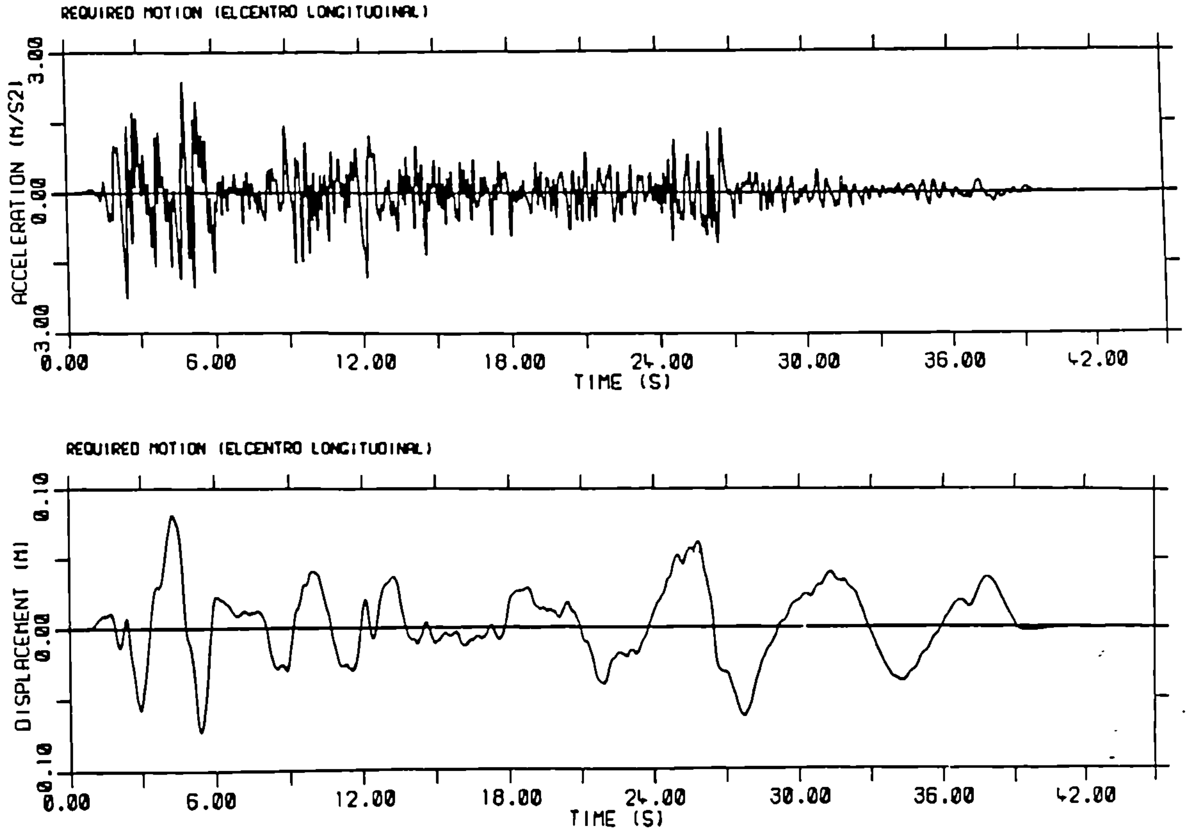


Fig. 4.15 The 1940 N-S component of the El Centro time history

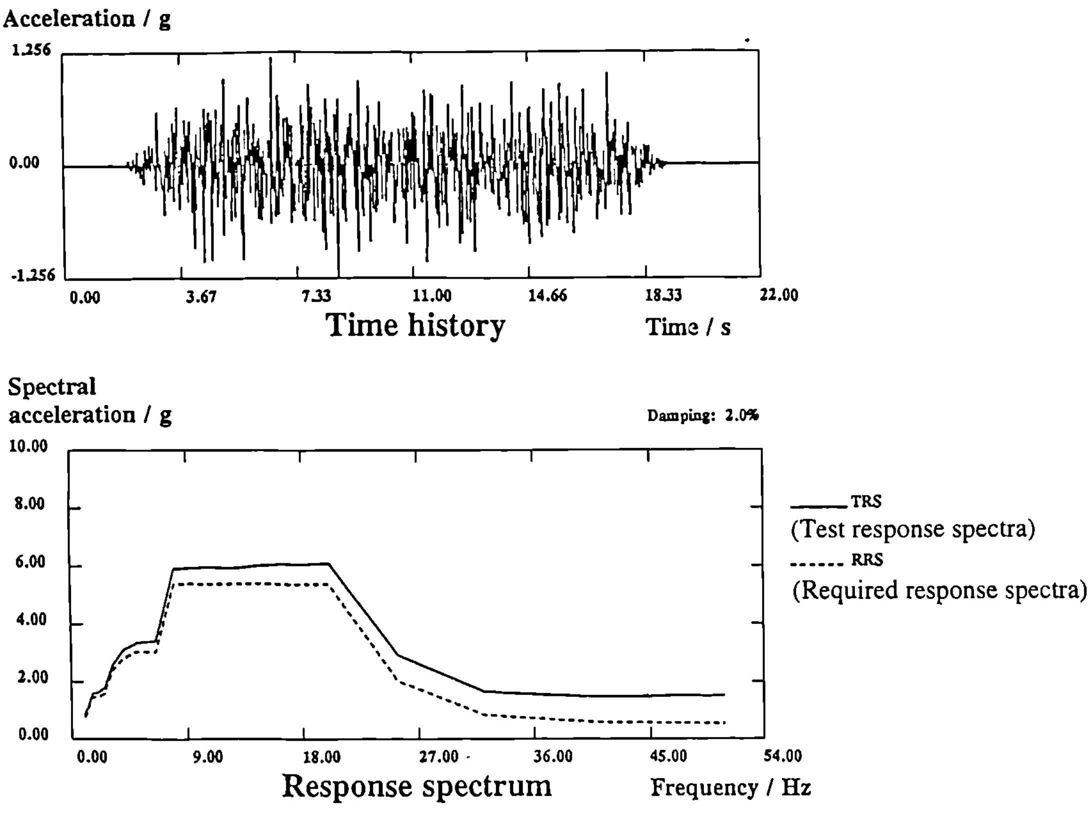


Fig. 4.16 Artificial earthquake and the spectrum used to generate it

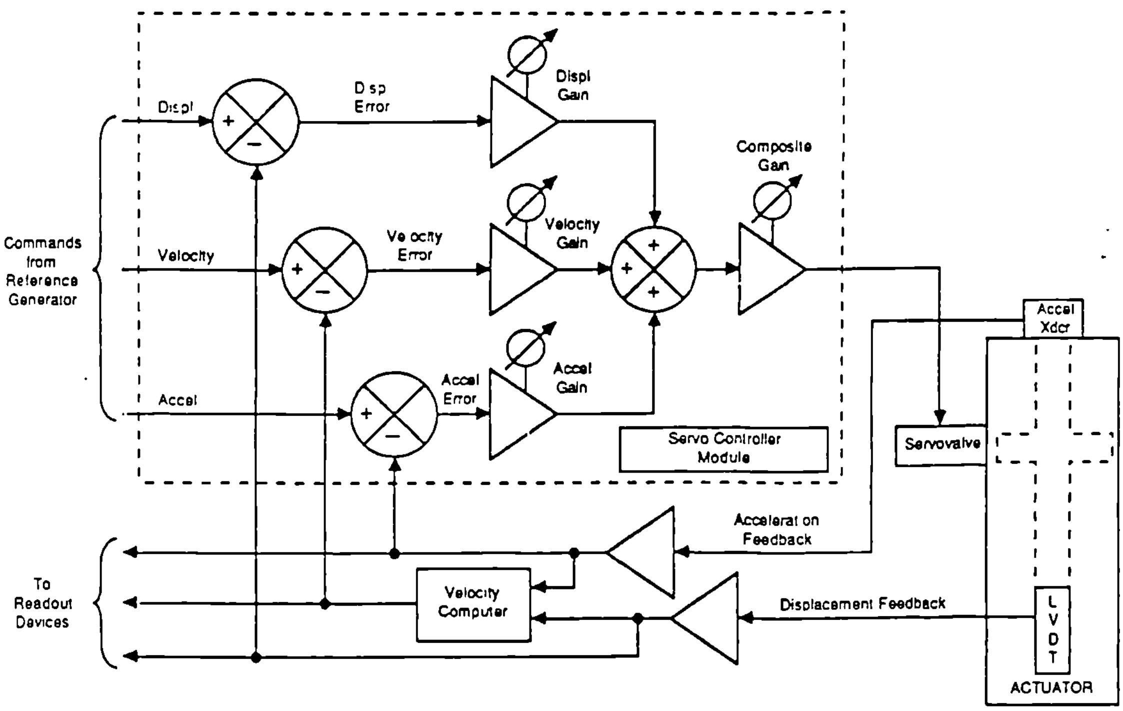


Fig. 4.17 Diagram of hardware feedback loops used in MTS Ltd. shaking tables

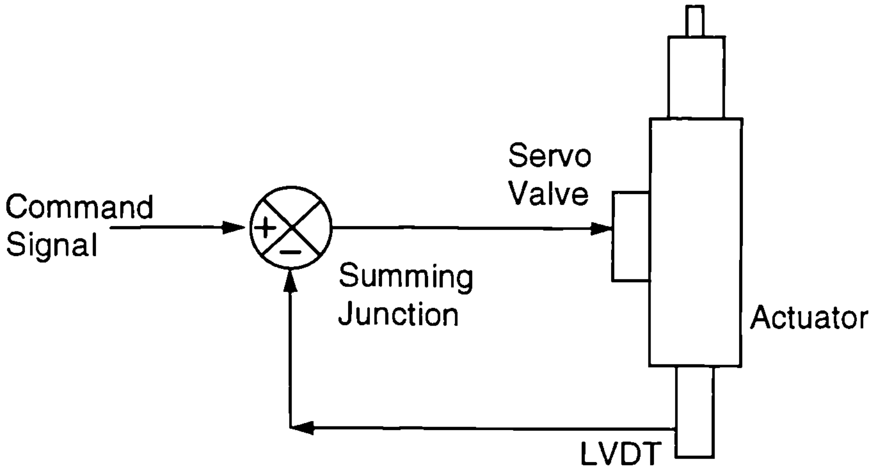


Fig. 4.18 Diagram of proportional feedback loop

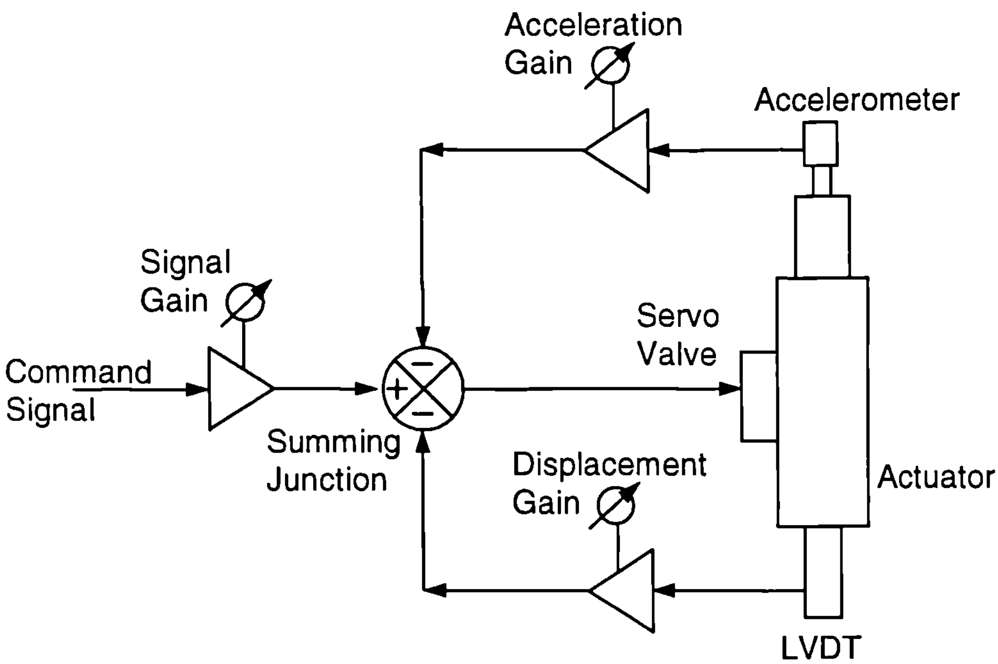


Fig. 4.19 Diagram of analogue hardware feedback loops in the Bristol table

(details of the abbreviations on these figures can be found in Appendix B.2).

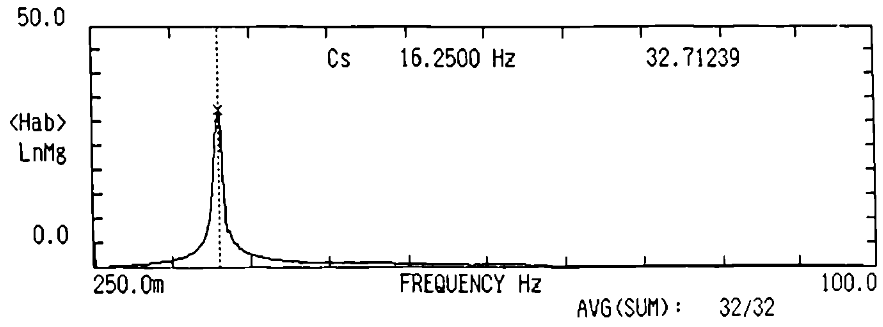


Fig. 4.20 Typical system transfer function at Bristol before tuning

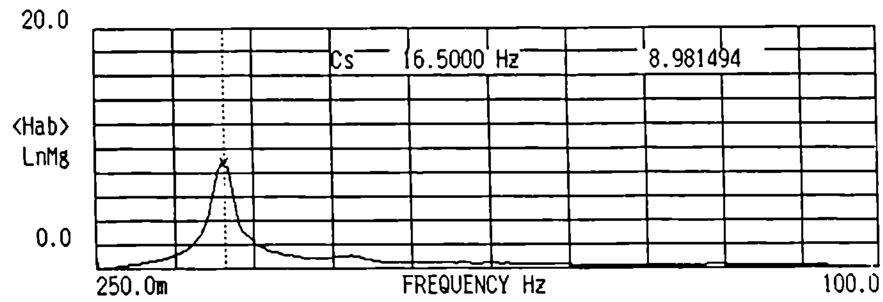


Fig. 4.21 Typical system transfer function at Bristol after tuning

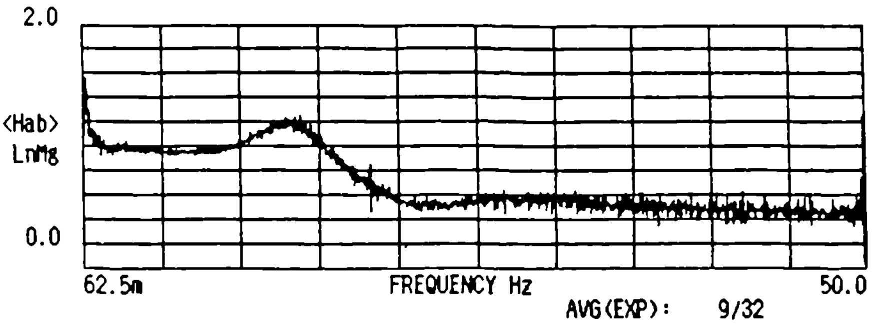


Fig. 4.22 Typical system transfer function at Athens before tuning

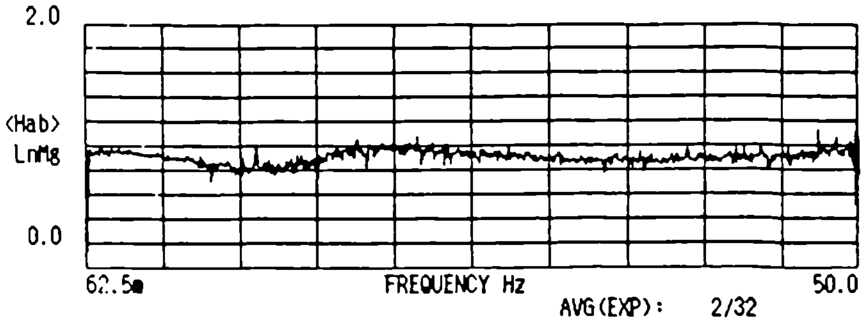


Fig. 4.23 Typical system transfer function at Athens after tuning

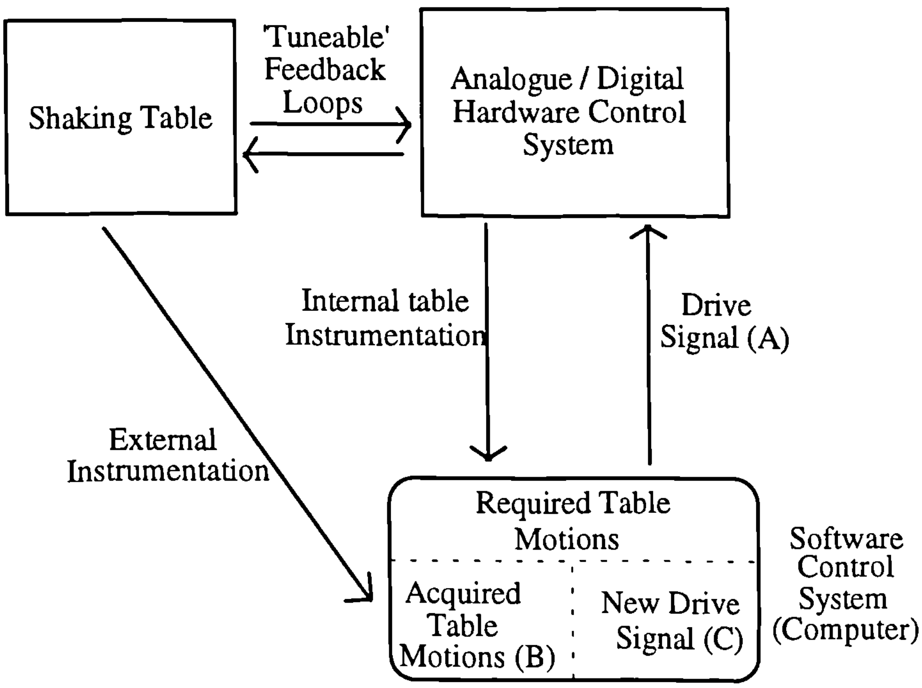


Fig. 4.24 Representation of typical software control system for a shaking table

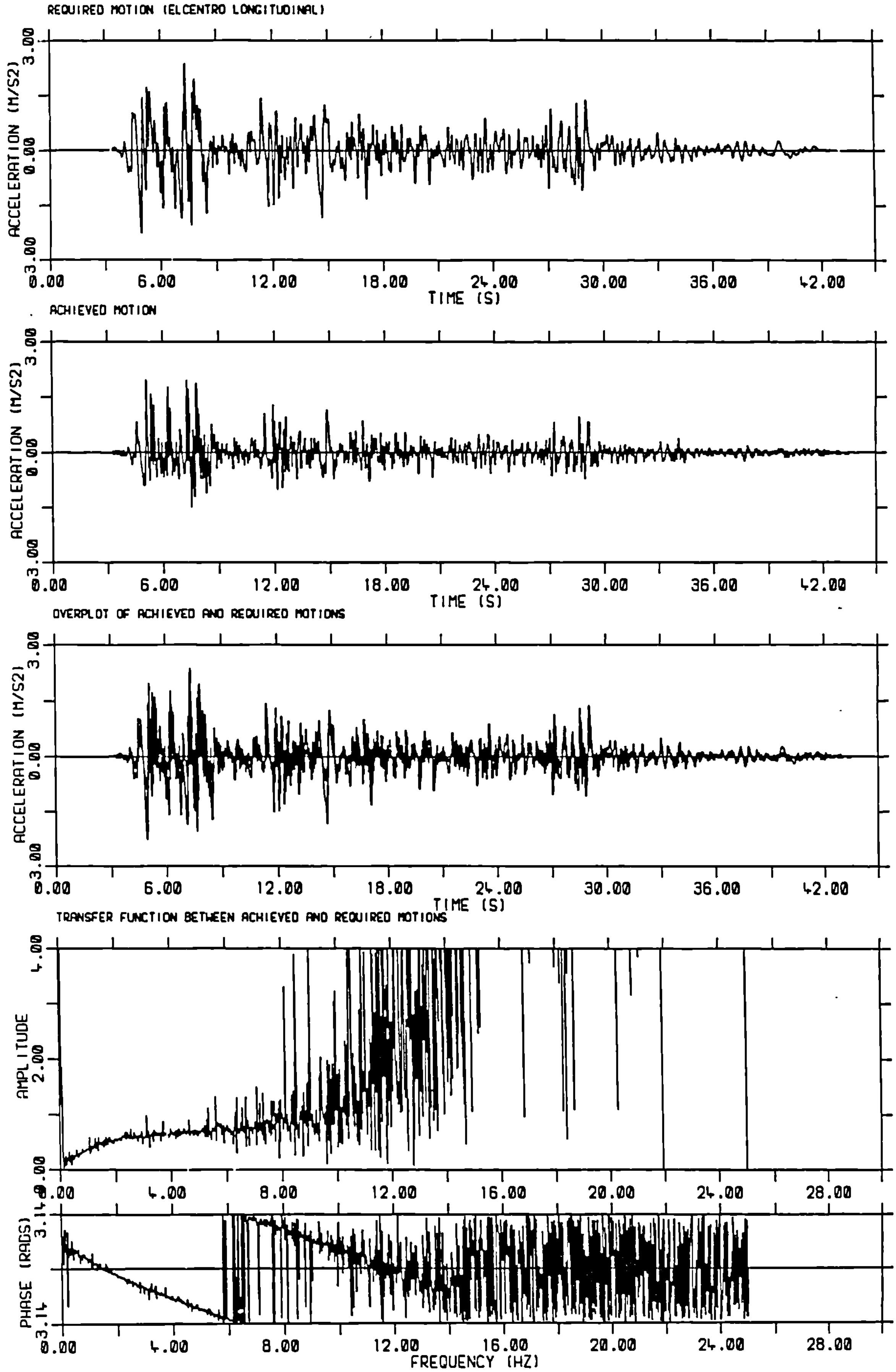


Fig. 4.25 Typical first attempt at reproducing a time history

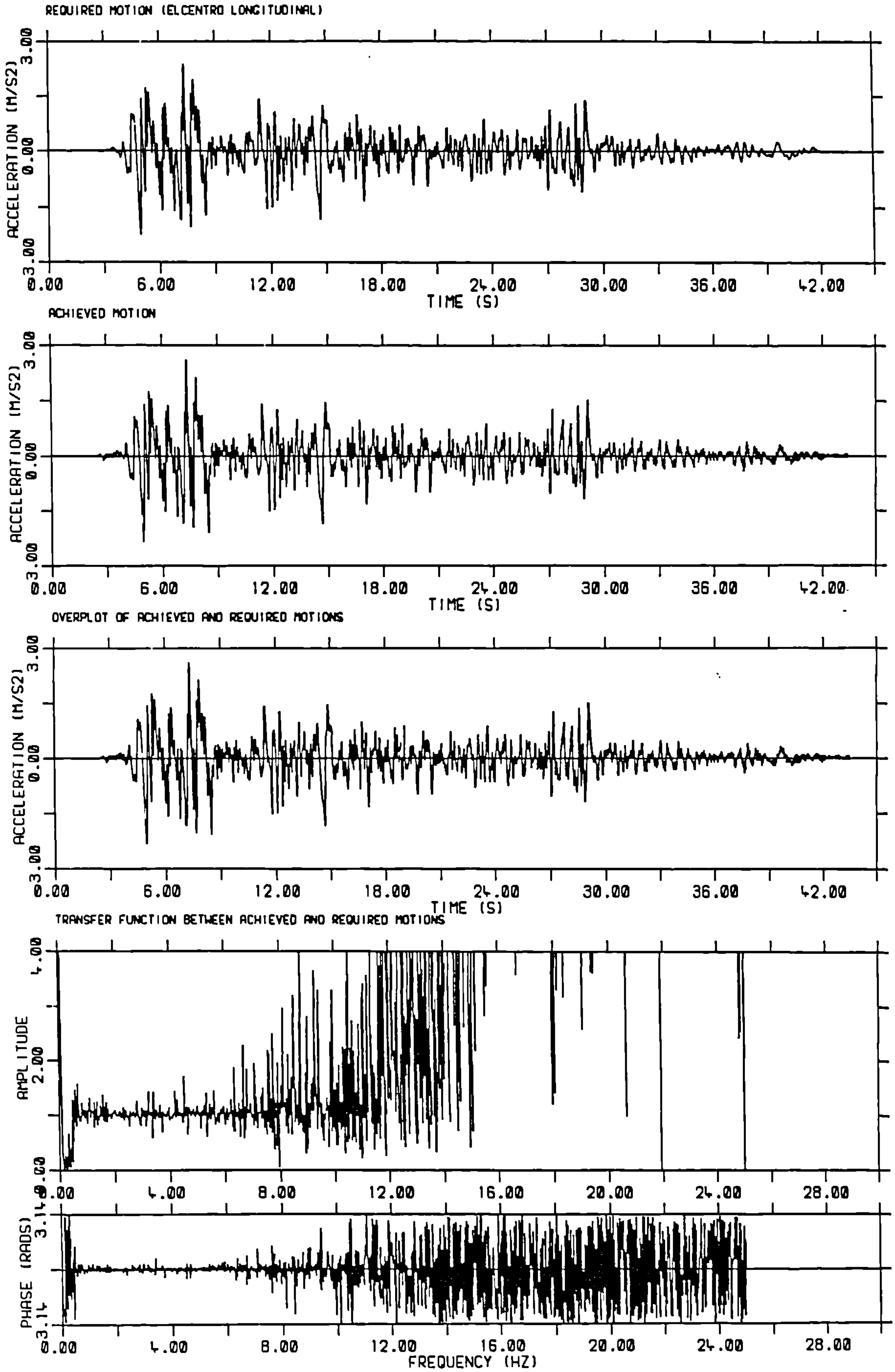


Fig. 4.26 Typical reproduction of a time history after three iterations of the matching process

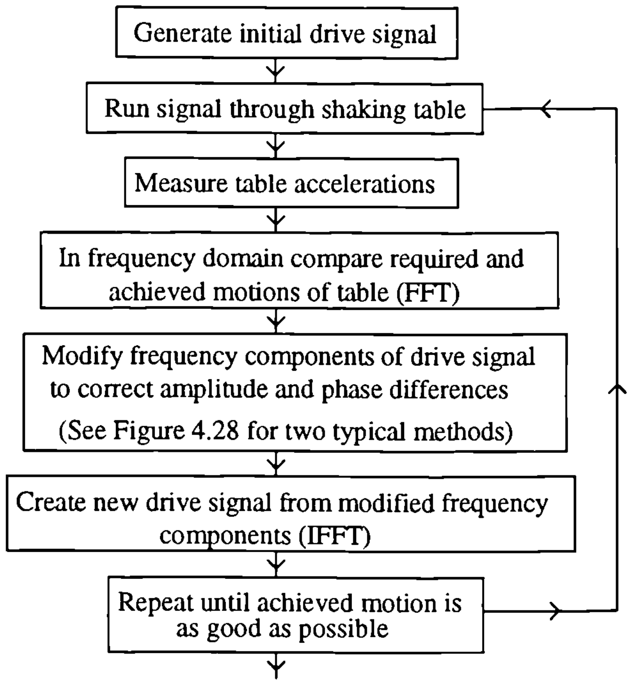


Fig. 4.27 Flow diagram for iterative time history matching

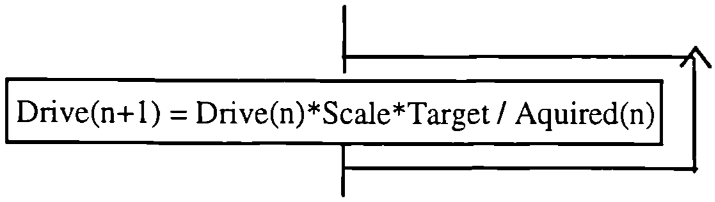


Fig. 4.28a Linear calculation of drive signals (in frequency domain) for iterative time history matching

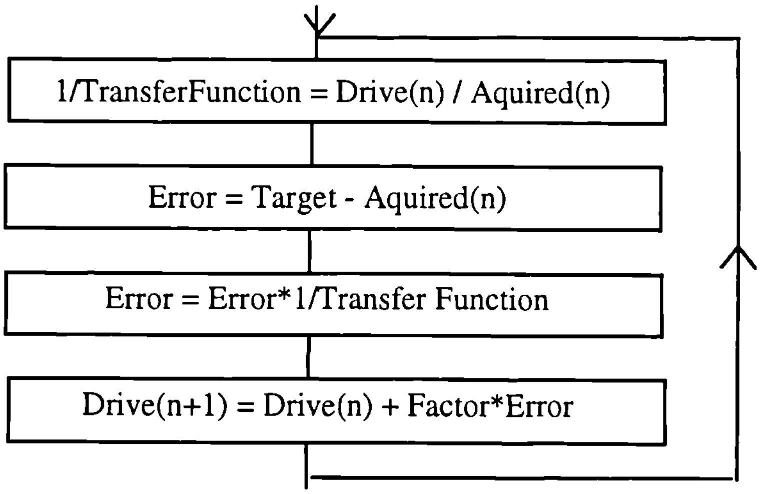


Fig. 4.28b Non-linear calculation of drive signals (in frequency domain) for iterative time history matching

Chapter 5

Assessing the Performance of Shaking Tables

5.1 Introduction

Before this research started, very little information was available concerning different kinds of shaking table facilities or giving guidelines for their use in research (Chapter 2). Little was known of the precise limitations of each type of test facility, and in addition it was difficult to define the limitations of any particular test facility in relation to a common standard, because the limitations of a facility generally were known only to one or two operators at the particular facility.

The few typical results presented in Chapter 4 show that it was possible, in some circumstances, to control the motion of a shaking table platform reasonably accurately and to compensate for any mechanical resonances and inadequacies in the hardware control systems using an appropriate iterative matching process. However, at the time this research programme began, it was clear that although good results could be achieved in these circumstances with an empty table, the results of the iterative matching process with a specimen on a table were generally not as good because of interaction between the specimen and the table. No-one had attempted to characterise and critically compare the performance of different shaking tables, nor to identify the most effective techniques used in operating the tables or to make a detailed assessment of the effect of any imperfections in control on the tests being performed.

It was evident, therefore, that there was significant scope for improving our knowledge of shaking table behaviour, for finding a method of directly comparing the results from different tables, and for the development of new techniques that could improve the way in which shaking tables are used in research.

The work described in this chapter was carried out in collaboration with colleagues at the Laboratory for Earthquake Engineering (LEE) at the National Technical University of

Athens, the Structural Dynamic Testing Laboratory of ISMES, Seriate, Italy, and the Laboratorió Nacional De Engenharia Civil (LNEC), Lisbon as part of the EU funded standardisation project. Information is drawn from the final report on this project (Crewe, 1997) for which the author was the editor and primary contributor. The interpretations and discussions of the results of this project, as presented in this chapter, are those of the author.

5.2 The European Consortium Of Earthquake Shaking Tables (ECOEST)

Many shaking tables throughout the world are being used to study the dynamic effects of earthquakes on structures. Each of these tables has its own performance characteristics. However, as every experimentalist knows, different pieces of equipment designed to carry out the same experiment cannot be guaranteed to yield the same results. This is particularly the case for complex equipment, such as shaking tables. If, therefore, direct comparisons of results from different shaking tables are to be meaningful, it is essential that the potential effects of the performance characteristics of each table are well understood.

A principal aim of this research, therefore, was to explore methodologies for a comparison of shaking tables with a view to establishing standardised benchmark tests that could be used to validate and hence improve shaking table usage. The four organisations with shaking tables that form the European Consortium Of Earthquake Shaking Tables (ECOEST) were well suited to such a study, as their tables had a relatively small range of payload capacities but had very different hardware control systems, software control systems and performance characteristics.

5.3 The four shaking tables studied

The four ECOEST shaking tables are all medium sized facilities (10 to 100 tonne capacity). As can be see from table 5.1, none of these four shaking tables can be considered to be exceptionally large (capacity > 100 tonnes); but they nevertheless all have significant capacities and they are all used extensively for full scale and model scale testing. Brief descriptions of each of the four shaking tables studied are given below. More detailed descriptions of the facilities at each site may be found in the "Standardisation of Shaking Tables" report (Crewe, 1997). It should be noted that the software and hardware descriptions are as existed between 1993 and 1996 when the majority of testing at the four sites was performed. These descriptions will become outdated when any subsequent upgrades take place at the four sites.

The Athens, Bristol and ISMES shaking tables have full six-axis control, while the LNEC table is mechanically restrained to move only in the translational directions. The differing performance characteristics of the four tables mean that they are generally used for different types of research. The Athens and LNEC tables, having a relatively small operating frequency range, are most suited to larger scale concrete and masonry testing. The Bristol and ISMES tables having higher frequency ranges are more suited to performing model testing where the scaling laws (table 3.1) require that the scaled time history has higher frequencies than the original signal. However all the tables, and even the LNEC table with its large capacity and torque tube restraining system designed specifically for the large scale testing of masonry structures, are used perform a wide variety of tests.

Table 5.1 Summary of performance characteristics of the four shaking tables.

	Platform Mass (tonnes)	**Table Capacity (tonnes) ***	**Max. horiz. accel. Bare table (g) ***	**Max. operating frequency (Hz) ***
Athens	10	10	2.0	25
Bristol	3	15	4.5	100
ISMES	11	30	3.0	120
LNEC	40	40	1.1 - 1.8 (axis dependent)	20

* The capacities and performances quoted are as at September 1998. These performances may change with any subsequent upgrading of the shaking tables.

The naming convention that is used throughout this dissertation to define the six possible degrees of freedom of the shaking table platform motion is shown in figure 5.1. Motion can occur in the following six axes: X (Longitudinal translation), Y (Lateral translation), Z (Vertical translation), R (Roll ≡ rotation about X axis), P (Pitch ≡ rotation about Y axis), W (Yaw ≡ rotation about Z axis).

5.3.1 The National Technical University of Athens shaking table

The shaking table at NTUA (figure 5.2) was commissioned in 1985 after a 4-year construction period, and is housed in the large, purpose built Laboratory for Earthquake Engineering. The table has six degree-of-freedom control, offering control of the three orthogonal translational degrees of freedom (i.e. two horizontal and the vertical) and the associated rotational degrees of freedom (i.e. roll, pitch and yaw). The steel platform measures 4 m by 4 m and weighs 10 tonnes. It can carry a maximum payload of 10 tonnes. The platform is driven by eight 160 kN servo-hydraulic actuators. Four act horizontally and four act vertically at the corners of the platform. The servo-hydraulic, hardware and software control systems were manufactured by MTS Inc., from the USA.

Table 5.2 Performance characteristics of the Athens table.

Table size	4 m by 4 m
Platform mass	10 tonnes
Maximum specimen mass	10 tonnes
Maximum specimen height	11 m
Controlled degrees of freedom	6
Translation	X, Y, Z
Rotation	θ_x, θ_y, θ_z
Longitudinal (X) or Lateral (Y)	
Displacement	± 100 mm
Velocity	± 1000 mm/s
Acceleration	± 2g (zero payload)
Vertical (Z)	
Displacement	± 100 mm
Velocity	± 1000 mm/s
Acceleration	± 4g (zero payload)
Frequency Range	0 - 25 Hz

The hardware control of the shaking table is provided by an MTS analogue system offering extensive control over various feedback loops. The software control and data acquisition systems are based on a PDP11/34 minicomputer running under the RSX11M operating system. A wide range of earthquake, sinusoidal, random and other signal forms may be applied. Earthquake time histories are iteratively matched using an out-of-real time, adaptive control algorithm.

5.3.2 The Bristol University shaking table

The shaking table at Bristol University (figure 5.3) is housed in the Earthquake Engineering Research Centre. It was funded originally, in 1985, by the UK Science and Engineering Research Council (SERC) and Bristol University at a cost of over £1.0m. Subsequently, the Research Council was renamed the Engineering and Physical Sciences Research Council (EPSRC), and the shaking table is now officially known as the EPSRC Earthquake Simulator. The whole system was designed and built in-house, in collaboration with Silveridge Technology Ltd., a UK company.

Table 5.3 Performance characteristics of the Bristol table.

Table size	3 m by 3 m
Platform mass	3 tonnes
Maximum specimen mass	15 tonnes
Maximum specimen height	4 m
Controlled degrees of freedom	6
Translation	X, Y, Z
Rotation	θ_x, θ_y, θ_z
Longitudinal (X) or Lateral (Y)	
Displacement	± 150 mm
Velocity	± 700 mm/s
Acceleration	± 4.5g (zero payload)
Vertical (Z)	
Displacement	± 150 mm
Velocity	± 700 mm/s
Acceleration	± 7g (zero payload)
Frequency Range	0 - 100 Hz

The Bristol shaking table has six degree-of-freedom control, giving control over the three orthogonal translational degrees of freedom (i.e. two horizontal and the vertical) and the associated rotational degrees of freedom (i.e. roll, pitch and yaw). The cast aluminium platform, measures 3 m by 3 m in plan and weighs 3 tonnes. The platform is in the form of an inverted pyramid, with a depth of over 1 m at the centre, and is stiffened internally by a honeycomb of diaphragms that give it a very high bending stiffness. Its first flexural natural frequency is over 100 Hz, well beyond the seismic test range and rendering it effectively rigid. The platform can carry a maximum payload of 15 tonnes.

The platform is driven by eight, 50 kN servo-hydraulic actuators. Four act horizontally and are arranged around the perimeter of the platform. A further four act vertically at the corners of the platform.

At the start of these investigations (Nov. 1993), the shaking table was controlled via an analogue servo-control panel and a 486 66 MHz processor Personal Computer. Towards the end of this research the hardware control system was updated (early 1997) by replacing the old analogue controller with a DARTEC 9600 digital controller which is controlled via a Pentium 233 MHz processor PC. This PC also provides the software control that allows a wide range of earthquake, sinusoidal, random and other signal forms to be applied to the table. Earthquake time histories are iteratively matched out-of-real time using a frequency-domain based adaptive control algorithm. Data can be collected simultaneously on up to 64 channels on a separate data acquisition computer that is synchronised to the control computer.

All research and commercial testing in the EERC is managed under a Quality Assurance system that complies with British and ISO standards.

5.3.3 The ISMES MASTER shaking table

The MASTER shaking table at ISMES is located in the Structural Dynamics Testing Laboratory (figure 5.4). It was commissioned in 1984 and is housed in a large, purpose built laboratory. The ISMES MASTER shaking table has six degree-of-freedom control (i.e. two horizontal and vertical plus the roll, pitch and yaw motions). The steel platform measures 4 m by 4 m and weighs 11 tonnes. It can carry a maximum payload of 30 tonnes.

The platform is driven by four 150 kN vertical servo-hydraulic actuators and four 250 kN horizontal servo-hydraulic actuators at the corners of the platform. The servo-hydraulic and analogue control systems were manufactured by MTS Inc. The new, recently installed, digital control system was developed by I*STAR, which has recently been incorporated into the Concurrent Computer Corporation, Paris.

Table 5.4 Performance characteristics of the ISMES table.

Table size	4 m by 4 m
Platform mass	11 tonnes
Maximum specimen mass	30 tonnes
Controlled degrees of freedom	6
Translation	X, Y, Z
Rotation	θ_x, θ_y, θ_z
Longitudinal (X) or Lateral (Y)	
Displacement	± 100 mm
Velocity	± 550 mm/s
Acceleration	± 3g (zero payload)
Vertical (Z)	
Displacement	± 100 mm
Velocity	± 440 mm/s
Acceleration	± 2g (zero payload)
Frequency Range	0 - 120 Hz

The shaking table is controlled by an MTS analogue system which provides the basic hardware control for the table. The software control system of the MASTER shaking table is called NewMACS (New Multi Axes Control System). Various software packages within this system can be used to produce earthquake, sinusoidal, random and other signal forms. This software runs on a Concurrent ISCV-MC554 (7250) computer with three 33 MHz 68040 co-processors and 32 Mb of RAM memory. Earthquake time histories are iteratively matched using an out-of-real time, adaptive control algorithm which gradually improves the platform motion during multiple tests.

5.3.4 The 3D LNEC shaking table

The 3D shaking table at LNEC is located in the Centre for Studies and Equipment in Earthquake Engineering (figure 5.5). It was commissioned in 1995 and is housed in a large, purpose built laboratory. The LNEC shaking table has three degree-of-freedom control giving control of the three orthogonal translational degrees of freedom (i.e. two horizontal and the vertical). The three rotational degrees of freedom (i.e. roll, pitch and yaw) are constrained by a set of torque tubes. Figure 4.3 shows the arrangement of the torque tubes in the LNEC table. The steel platform measures 5.6 m by 4.6 m and weighs 40 tonnes. It can carry a maximum payload of 40 tonnes. The platform is driven by one 1000 kN longitudinal, two 300 kN lateral and one 300 kN vertical servo-hydraulic actuators situated on the centrelines of the platform. The servo-hydraulic, analogue and digital control systems were manufactured by INSTRON.

Table 5.5 Performance characteristics of the LNEC table.

Table size	5.6 m by 4.6 m
Platform mass	40 tonnes
Maximum specimen mass	40 tonnes
Controlled degrees of freedom	3
Translation	X, Y, Z
Longitudinal (X)	
Displacement	± 175 mm
Velocity	± 200 mm/s
Acceleration	± 1.8g (zero payload)
Lateral (Y)	
Displacement	± 175 mm
Velocity	± 200 mm/s
Acceleration	± 1.1g (zero payload)
Vertical (Z)	
Displacement	± 175 mm
Velocity	± 200 mm/s
Acceleration	± 0.6g (zero payload)
Frequency Range	0 - 20 Hz

The hardware control of the shaking table is provided by an INSTRON 8580 digital control system with the control and iterative matching software running on two independent computers. INSTRON RSPlus software provides control over the shaking table hardware and the SPiDAR matching software generates the signal forms required for the tests. These two computers are linked via a network. Both computers are 486 66 MHz processor PCs but while PSPlus runs under MS DOS, the SPiDAR software runs under a UNIX operating system. Earthquake time histories are iteratively matched using an out-of-real time, adaptive control algorithm which uses a non-square matrix approach which takes into account the frequency range over which each type of transducer is most accurate.

5.4 Strategy for comparison of shaking tables

5.4.1 Methodology for shaking table evaluation

The investigations at the four laboratories into the use of shaking table facilities for earthquake engineering research started in 1993 and were separated into three phases; a software review, an operations review and a performance review. The principal aims of the software review were to validate the existing control, data acquisition and data processing software at the four sites, and to compare and contrast the techniques embodied in the software. In the operations review, the procedures adopted at each site for maintenance of the shaking tables, and the planning and execution of the tests, were compared. The primary objective of this review was to bring together the best practices from each site with the aim of enhancing the quality of research at all sites. A great deal was learnt from these two reviews but these results fall outside the scope of this dissertation. Further details can be found in "Standardisation of Shaking Tables" (Crewe, 1997).

The performance review formed the dominant part of this research. It consisted of a series of tests on each shaking table using identical rigid and flexible payloads. The main aim of these tests was to characterise the dynamic performance of each table in a common way. In particular, the frequency response and the ability of each shaking table to accurately reproduce several different time histories was investigated. A flexible, single degree-of-freedom specimen was designed such that its natural frequency, mass and centre of gravity could be adjusted over the typical operating ranges of the four facilities. Two identical specimens were fabricated in Bristol, and one was shipped to Athens, then to ISMES, and finally to LNEC as the tests progressed.

5.4.2 Aims and objectives of the test programme

The major part of this research concentrated on the comparison of the performances of the four shaking tables. The principal aim was to determine whether each shaking table could reproduce the same platform motions for specified tests. A further aim was to produce a detailed characterisation of the frequency response of each table, which could be used as a benchmark against which the performance of the table could be measured in the future, for example after major upgrades of hardware and software or during regular maintenance. An additional benefit of the characterisation studies was the identification of limitations or features of the table performance that might affect the viability of an experiment. Such knowledge is essential in the design and planning of a test.

The performance of any shaking table is dependent on the nature of the payload being shaken. In particular, it will be affected by the mass, centre of gravity and flexibility of the payload. From the point of view of control, the shaking table and test specimen must be viewed as one complete system whose frequency response will change with each specimen.

The objective of many experiments is to observe the mode of failure of a test specimen. The resulting non-linear behaviour of the specimen will lead to changes in the table-specimen frequency response during (say) a simulated earthquake, which the table control system should, ideally, be able to cater for while maintaining the fidelity of the required platform motion. A major objective of the research, therefore, was to design an adaptable test specimen with the facility to vary its mass, centre of gravity and flexibility, and hence its natural frequency. This proved to be a difficult task. An evaluation of the test specimen is given in §5.4.4. Due to the constraints with timetabling of the tests at the four laboratories it was necessary to construct two identical test specimens which were shipped between the four sites as the test programme progressed.

5.4.3 Scope of the test programme

The test programme was divided into two main phases. In the first phase, the frequency response of each shaking table was characterised for a range of payloads. The frequency response function of each axis was measured individually, and cross-coupling between various axes was also measured. This allowed the natural frequencies of the shaking tables to be rigorously explored and the effect of control system tuning to be evaluated. In addition, the effects of specimen interaction on the response of the shaking table were identified. An extensive frequency response function database was established for each shaking table to provide a benchmark for future reviews (see Appendix A).

In the second phase of testing, the ability of the shaking tables to recreate specified ground motions was evaluated. Two different recorded earthquakes were chosen. The classical 1940 N-S El Centro (California) record (see the upper plot of figure 5.12) was used to check the similarity of response in the X, Y and Z translational axes individually. This record is typical of a long duration earthquake. It lasts for 40 seconds, reaches peak accelerations of about 0.3g and has strong low frequency components that lead to large displacements. The record was chosen partly because it places long duration power and oil flow demands on the shaking table hydraulics. The second earthquake chosen was the 1989 Kalamata (Greece) three-component record (one axis of which is shown in the upper plot of figure 5.14). This earthquake is typical of European events, having about 12 seconds of strong shaking, higher frequency components than the El Centro record, and reaches peak accelerations of about 0.3g. The uniformity of the system gains of the shaking tables was investigated by repeating the earthquakes at different peak acceleration levels. The fidelity of the table motions was checked by comparing the achieved acceleration and displacement time histories and response spectra with those derived directly from the target motions.

5.4.4 Design of the test specimen

The design of the test specimen proved to be a challenge. At low frequencies the specimen had to be very flexible and able to sustain large deflections if adequate platform accelerations were to be achieved. Such flexibility must, ideally, be achieved for a large mass. The strength requirements of a specimen in such conditions are considerable and demand relatively large structural sections that inevitably have a high stiffness. The combination of high flexibility (or low stiffness) with high strength is difficult to obtain. Many test specimen concepts were evaluated before a relative simple, single degree-of-freedom structure was chosen (figure 5.6). The specimen consisted of a variable mass supported on four steel columns. The total mass of the specimen could be adjusted from 2 tonnes to 8 tonnes by adding 1 tonne steel blocks. The centre of gravity of the mass could also be adjusted from about 1 m to 2 m above the shaking table platform. The design also allowed the steel blocks to be used as rigid payloads, while the column sections could be readily changed in size and number if required in the future. The general arrangement of the flexible test specimen is shown in figure 5.7. Calculations for the predicted major axis natural frequencies of this structure in various arrangements are shown in table 5.6. The structure was assumed to be a single degree-of-freedom lumped-mass oscillator for the purposes of these calculations.

Table 5.6 Fundamental frequencies for various arrangements of the flexible model.

Number of Masses	**Equivalent lumped mass (kg)**	**Major axis Fundamental frequency of the model for various heights of the equivalent lumped mass (Hz)**					
		1.0 m	1.5 m	1.75 m	1.85 m	1.95 m	2.0 m
Columns only	380	68.34	37.20	29.52	27.16	25.10	24.16
Cols. + 1 mass	1455	34.92	19.01	15.09	13.88	12.83	12.35
Cols. + 2 masses	2530	26.49	14.42	11.44	10.53	9.73	9.36
Cols. + 3 masses	3605	22.19	12.08	9.58	8.82	8.15	7.84
Cols. + 4 masses	4680	19.47	10.60	8.41	7.74	7.15	6.88
Cols. + 5 masses	5755	17.56	9.56	7.59	6.98	6.45	6.21
Cols. + 6 masses	6830	16.12	8.77	6.96	6.41	5.92	5.70

The specimen did not perform quite as predicted, and the natural frequencies measured were about 10% lower than those calculated. This variation in the performance of the test structure occurred because the connections between the columns and the steel slabs at the top or base of the structure were not completely rigid, as was assumed in the original design. The fact that the structure did not behave quite as predicted did not affect the tests, and the greater flexibility of the structure allowed it to be tested with higher platform accelerations.

The specimen was designed to have a maximum mass of about 8 tonnes which is just below the maximum capacity of the Athens table. This would allow the specimen to test the Athens table practically up to its payload limit. The shaking tables at the other three sites have larger capacities but this model was still able provide a significant test of the control systems on these tables. In addition to comparing the specimen mass with the table capacity, the ratio of the specimen-to-platform mass also needed to be considered. This was because a larger specimen-to-platform mass ratio is more likely to cause difficulties in the control of the platform motion (Takahashi et al., 1974). The worst cases for the four tables with the maximum specimen mass are shown in table 5.7. It can be seen that because the platform of the Bristol table is very light, with a platform mass of only 3 tonnes compared with a maximum specimen mass of 8 tonnes, it is more likely to experience more significant table-specimen interaction than the other three tables. The specimen-to-platform mass ratios that were used in the majority of the tests in each laboratory are also shown. By reducing the specimen masses used in the Athens and Bristol tests the ratios of specimen mass to platform mass were kept more similar, but the Bristol table with a mass ratio of 1.66 was still much more likely to be effected by the flexible model. The large payload at 166% of the platform mass had a significant effect on the response of the Bristol table (§5.5.1.1) and provided a good test of this table performance, although it was still not testing the table right to its limits.

Table 5.7 Specimen / platform mass ratios

Shaking table	**Platform mass (tonnes)**	**Specimen / platform mass ratio (maximum specimen mass = 8 tonnes)**	**Total specimen mass (in majority of tests)**	**Specimen / platform mass ratio (specimen mass shown in previous column)**
Athens	10	0.80	5 tonnes	0.50
Bristol	3	2.66	5 tonnes	1.66
ISMES	11	0.73	8 tonnes	0.73
LNEC	40	0.20	8 tonnes	0.20

However, while the small platform mass at Bristol is more likely to lead to increased table-specimen interaction, there are significant benefits in having a small platform mass if the table can be controlled in such a way as to minimise this interaction. The small platform mass at Bristol gives this table a very large dynamic range, and the actuators are able to accelerate the bare platform at up to 4.5g horizontally which is significantly higher than can be achieved on the other tables. The shaking tables at Athens, ISMES and LNEC have much greater platform masses of between 10 tonnes and 40 tonnes, with payload capacities between 10 tonnes and 40 tonnes. However, the actuators are only capable of accelerating these bare platforms at up to 3g horizontally. These performance parameters, summarised in table 5.1, meant that the flexible test specimen could test the performance of the smaller capacity Athens and Bristol tables almost to their limits, but was a little too light to test the ISMES and LNEC tables over their full payload range.

The specimen itself was tuneable (in use) to a broad range of frequencies between 6 Hz and 20 Hz. However, it was not possible to design a 'massive' specimen that had natural frequencies below 6 Hz because such a specimen is required to withstand large deflections when subjected to any useful acceleration level. These large deflections would generate large bending moments in the specimen, and the slender sections required to keep the natural frequency low could not carry these moments. 'Massive' yet flexible structures are very difficult to build as the requirements for carrying mass and being flexible are mutually exclusive. The specimens used in this research proved to be very robust while being flexible enough to effectively test the control systems of the four tables. Overall, it was a successful, adaptable design.

The solid 1 tonne sections of the specimen were initially used to test the performance of each of the tables with different static payloads of up to 8 tonnes. The specimen was then assembled so that it weighed approximately 5 tonnes and had natural frequencies of 8.0 Hz and 6.8 Hz in its major and minor axes respectively for the tests in Athens and Bristol. In the tests at ISMES and LNEC the specimen was assembled so that it weighed approximately 8 tonnes and had natural frequencies of 7.6 Hz and 6.4 Hz in its major and minor axes respectively. These specimen masses were chosen to keep the specimen-platform mass ratios as similar as possible across the tests at the four laboratories.

5.4.5 Test procedures

When this research started it was initially proposed that four different types of test should be performed in order to obtain a good comparison of shaking table performance. These would be:

- Impulse Signal Tests
- Random Signal Tests
- EC8 Response Spectra Tests
- Time History Tests - single and three-axis motions

These tests were designed to test the performance of a shaking table in several different ways. The test sequence performed on each table was designed firstly to characterise and understand the performance of each table in its own right. The impulse signal tests and the random signal tests were designed to identify the combined response of the control system and table, either to impulse excitation or to random signal excitation (0 - 100 Hz). Then, the ability of each table to reproduce several earthquake motions would be tested. These tests would ultimately aim to prove that identical tests could be performed on each of the shaking tables. The response spectra tests would determine the ability of the table control system to produce an arbitrary platform motion that enveloped a defined response spectrum. The time history tests would determine the ability of the table control system to produce a required motion on the platform. In addition to comparing the required and achieved motions of the platform, the response spectra for the required and achieved platform accelerations were also compared.

To this end, the test program performed on each shaking table had two distinct phases. In the first phase, frequency response tests were used to characterise the dynamic characteristics of each table under various loadings. These tests also examined the extent to which the hardware controllers could compensate for table-specimen interaction. The frequency response functions of the Bristol and Athens tables were measured in all six degrees of freedom for bare platform, 4 tonne and 8 tonne rigid payloads, and 4 tonne and 8 tonne flexible payloads which had frequencies of about 10 Hz and 6 Hz respectively (Carydis et al. 1994). A reduced number of similar tests were performed in LNEC and ISMES after it became clear what the critical tests were. The frequency response between the input driving signal and the platform acceleration was computed using an Advantest modal analyser, except for the tests in LNEC where in-house software was used. Broadband 0.1g random noise (up to 100 Hz) was played through each degree of freedom in turn. The frequency response was measured in the direction of excitation and in all other degrees of freedom to check for cross-coupling effects. Once the table hardware had been tuned at each load level to give the best possible system transfer function, the time history reproduction tests were performed.

This second phase of testing looked at the ability of each software control system to drive the platform so as to reproduce a predefined motion. Two earthquake records were used to test the control systems. The N-S El Centro record was used to check the similarity of each of the translational axes. The Kalamata earthquake record was then used to check the performance of the tables with a three-axis input. All the tests were performed with no payload on the platform, then four and/or eight tonne payloads were added to the platform to assess how the performance was affected by the load on the table. Finally a specimen with tuneable frequency response was mounted on the platform to assess how the control system could cope with table-specimen interaction.

As testing started, first of all on the Bristol table, it became clear that some of the planned tests were superfluous, in particular the impulse signal tests and the response spectra tests. Some impulse tests were performed at Bristol to check that the random noise tests produced identical answers, but these tests were not subsequently performed at any of the other sites. The response spectra tests were also eliminated from the test programme once it became clear that the tables were able to accurately reproduce both the displacements and accelerations of a specific time history in most circumstances. This was a more onerous requirement than just producing suitable acceleration responses on the platforms, so it was felt that the simpler tests could be eliminated from the test programme.

An additional significant change between the tests performed at Bristol and Athens and the tests at ISMES was the elimination of the single-axis shake tests, based on results obtained in Bristol and Athens. This further reduction in tests meant that the trials included only the random motions and the three-axis shakes, and this considerably simplified the testing at ISMES. However, a few single-axis time history tests were performed at LNEC as this research programme provided a good opportunity to explore in detail the performance of this table which had only just been commissioned as testing began. Definitions of the critical tests and of the final recommended procedure that should be used to assess the performance of a shaking table are described in detail in §5.5.6. This procedure is based on the results and experience gained throughout the testing at the four shaking table laboratories.

5.4.6 Identification of strengths and weaknesses in table performance

The ideal hardware control system for a shaking table would have unit gain and no phase difference between the drive signal and the platform response in each axis, across the full range of operating frequencies, both with and without any model on the platform. This would mean that any signal used to drive the table would be reproduced perfectly by the platform without the need for any additional software control system. In addition, there would ideally be no cross-coupling between the responses in any two axes, meaning that movement in each axis was entirely independent of movement in any other axis. Unfortunately, because a shaking table is a complicated system where something as small as a change in oil temperature can affect its dynamics (§4.3.2.6), this ideal will never be achievable. However, the closer the system transfer function is to unity at all frequencies with no phase error when there is any loading on the platform, the easier it will be for the software control system to compensate for any errors in the hardware control. The results of the frequency response tests on each table were therefore compared with this ideal situation, and gave a good indication of the mechanical characteristics and effectiveness of the hardware control systems in the tables.

The time history response tests, on the other hand, tested the ability of the software control systems to compensate for any errors in the hardware control systems. Ideally the platform response would exactly follow the desired time history motions in the excited axes, with no motion occurring in any of the other axes. Therefore, the platform motions would not only have the same peak displacements and peak accelerations as the desired time histories but would also reproduce the correct instantaneous displacements and accelerations at all times during the shake. Analysis of the ability of the software control systems to accurately reproduce the time histories was performed by calculating the transfer function between the desired and achieved signals. Perfect reproduction would then result in a transfer function that showed unit gains with no phase error at all frequencies.

5.4.7 Suitability of time histories and test sequence

Both the El Centro and Kalamata time histories are low frequency shakes with most of their energy in the frequencies below 8 Hz. Figures 5.8 and 5.9 show the cumulative power spectra for the Kalamata acceleration and the displacement time histories to frequencies above 15 Hz. The cumulative power spectra for the El Centro acceleration and displacement motions are similar to those of the Kalamata shake. The Athens and LNEC tables are only controlled to 25 Hz and 20 Hz respectively, and they have little dynamic capacity above these frequencies. The two shakes with frequency contents up to 15 Hz were therefore well-suited to the dynamic range of these tables. The dynamic responses of the shaking tables at Bristol and ISMES, on the other hand, are good up to 100 Hz and 120 Hz respectively, so higher frequency shakes would be necessary to test these tables up to their performance limits. Much of the testing carried out on these shaking tables is on scale models, and in these cases the frequencies in the applied time histories are scaled (table 3.1), with the frequency content shifted towards the higher frequencies. Therefore scaled versions of these shakes could be more appropriate for testing the performance of these tables under normal operating conditions. However, for this series of comparative tests it was felt that more useful information could be obtained if all the four tables were tested with identical earthquakes. For the assessment of a single table it would be more appropriate to use scaled time histories or other earthquake records that have a significant frequency content up to the maximum operating frequency of the table.

5.5 Comparison of the four shaking tables

The following five sub-sections detail the specific tests and conclusions that can be drawn for the tests performed on each of the four shaking tables studied. The test programme started at Bristol in November 1993 (§5.5.1), continued in Athens during December 1993 (§5.5.2), moved to ISMES in November 1995 (§5.5.3) and finally concluded with the tests on the LNEC table in February 1996 (§5.5.4). This programme of tests was then supplemented by a further series of tests performed on the Bristol table in July 1997 that looked in particular at some of the key issues that were highlighted by the tests performed at the four laboratories and compared the performance of the old analogue controller at Bristol with the new DARTEC 9600 digital controller that was installed at the end of the test programme (§5.5.5*)* It should be noted that many of the initial tests performed at Bristol were not repeated at the other sites as it was felt that no new conclusions would be drawn from such additional tests.

The author also like to note that the results from the Athens, ISMES and LNEC laboratories were obtained in collaboration with the operators at these three sites. In particular the author thanks G Franchioni and A Campos-Costa who performed most of the basic processing of the data obtained at the ISMES and LNEC laboratories.

5.5.1 Bristol site

5.5.1.1 Frequency response

The full series of frequency response test results (see Appendix A) from the testing performed in November 1993 for the Bristol table have already been published and can be found in the "Athens/Bristol Shaking Table Assessment Project - Final Technical Report" (Crewe & Taylor, 1994).

These frequency response tests highlighted the fact that there are significant resonance regions in the frequency response of the Bristol shaking table. In the worst case an amplification in the platform response to an input signal at 16.25 Hz of 33.7 was recorded. This was caused by oil column resonance in the X-axis translational direction. The natural frequencies of the control system and the table were identified, and the resonances in all axes with their amplification factors are listed in table 5.8. This table includes values in the non-excited axes, and shows any cross-coupling that was present. The cross-coupling effects in the three main axis directions were very small, while those in the rotational axes were slightly larger. The cross-coupling of the rotational axes was probably caused by inadequate tuning of the old analogue control hardware and occurred at all the relevant oil column resonances. The values listed in table 5.8 are the bare platform responses, with no additional payload, to an impulse input on only one axis at a time.

Table 5.8 Resonance frequencies and amplification factors of the Bristol table with single-axis impulse excitation before tuning

Excitation	**Response Axis**											
Axis	**X (Longitudinal)**		**Y (Lateral)**		**Z (Vertical)**		**R (Roll)**		**P (Pitch)**		**W (Yaw)**	
	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification
X (Longitudinal)	**16.25**	**33.71**	17.0	0.23	16.0	0.46	31.5	0.19	16.0	0.72	35.2	1.66
Y (Lateral)	17.00	0.31	**16.25**	**31.73**	16.5	0.94	16.0	0.31	16.5	0.20	33.25	2.80
Z (Vertical)	23.5	2.05	22.75	1.63	**23.0**	**22.34**	22.75	1.34	23.5	1.66	22.75	0.37
R (Roll)							**30.0**	**16.05**				
P (Pitch)									**30.0**	**14.95**		
W (Yaw)											**34.75**	**31.31**

Table 5.9 Resonance frequencies and amplification factors of the Bristol table with single-axis impulse excitation after tuning

Excitation	**Response Axis**											
Axis	**X (Longitudinal)**		**Y (Lateral)**		**Z (Vertical)**		**R (Roll)**		**P (Pitch)**		**W (Yaw)**	
	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification
X (Longitudinal)	**16.5**	**8.98**										
Y (Lateral)			**15.75**	**5.9847**								
Z (Vertical)					**22.75**	**14.22**						

By subsequently making minor adjustments to the feedback loop gains in the analogue control hardware it was possible to reduce the amplification factors at resonance to between 20% and 50% of their previous values (see table 5.9). The effects of tuning on the system resonances, natural frequencies and system characteristics of the Bristol table are now much better understood; and had they been so well understood in 1993, it would have been possible further to improve the tuning of the control system for the Bristol table. Subsequent comparison of the frequency response results from Bristol with those obtained later from the other tables (§5.5.2.1, §5.5.3.1, §5.5.4.1) shows that a system transfer function close to unity at all frequencies can be obtained in some cases. However, the control hardware in Bristol (when these tests were performed) was not as sophisticated as the MTS or INSTRON systems, and it would not have been possible to achieve this accuracy of tuning with the existing hardware. The MTS hardware, for example, enables the gains and phases of the frequency bandwidths associated with displacement (low frequencies), velocity (intermediate frequencies) and acceleration (high frequencies) to be tuned independently. This gives greater control over local system resonances and leads to almost-flat system transfer functions. The Bristol control hardware does not have this facility; instead, all frequencies are tuned together and the software control system deals (usually very effectively) with system resonances during seismic tests. A full investigation of the effects of adjusting different hardware control loops in the Bristol hardware was not possible within the time allotted for this research, and in any case has now become irrelevant with the addition of the new digital hardware controller. This controller, in conjunction with some new active-control software currently being developed, effectively adjusts all the feedback gains in the control system in real time. Further details of this development in shaking table control can be found in §6.7.

The frequency response of the bare platform measured at several increasing levels of input signal amplitude showed that the platform response is linear so long as the driving signal remains within the performance limits of the shaking table characteristics. The performance limits of the table being:

Vertical acceleration (no payload):	5.6 g
Vertical acceleration (8 tonne payload):	1.2 g
Vertical velocity:	0.7 m/s
Vertical displacement:	± 150 mm
Horizontal acceleration (no payload):	4.7 g
Horizontal acceleration (8 tonne payload):	1.6 g
Horizontal velocity:	0.7 m/s
Horizontal displacement:	± 150 mm

Provided that the driving signal demanded a response that was within all these parameters, an increase in signal amplitude increased the platform response equally at all frequencies. By measuring the bare platform response at several input signal amplitudes the platform response was also found to be directly proportional to the input signal. Thus, the system gain was found to be uniform over all frequencies and could therefore be calibrated.

The series of tests that measured the frequency response of the table to a constant input signal with different payloads on the table showed that as the payload increased the frequency of the resonances in the system decreased and the damping for these regions of resonance increased. The frequency response of the control system and the table with the different payloads was investigated, and the resonances with their amplification factors, for those axes measured, are listed in table 5.10.

Table 5.10 Resonance frequencies and amplification factors for single-axis impulse excitation with various payloads

Excitation and Response Axis	**Payload**					
	0 tonnes		**4 tonnes**		**8 tonnes**	
	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification
X (Longitudinal)	**16.5**	**9.0**	**12.25**	**11.11**	**10.25**	**7.45**
Y (Lateral)	**15.75**	**6.0**	**11.25**	**7.64**	**9.5**	**6.21**
Z (Vertical)	**22.75**	**14.2**	**17.5**	**10.3**	**14.25**	**6.1**
R (Roll)	**30.0**	**16.0**	**21.5**	**12.2**	**17.75**	**9.3**
P (Pitch)	**30.0**	**15.0**	**21.5**	**12.6**	**17.75**	**9.0**
W (Yaw)	**34.75**	**31.3**	**24**	**20.5**	**19.5**	**14.7**

The values shown in table 5.10 are the platform responses, with a 0, 4 or 8 tonne rigid payload, to an impulse signal sent to one axis at a time. The resonances are mainly caused by the compliances in the actuator oil columns (§4.3.2.6). The results for this series of tests suggest that it could be possible to improve the overall performance of the table in certain circumstances by adding mass to the platform. This would reduce the system resonances, and so less software compensation would be required to cope with these resonances. However, the quantity of additional mass that can be added to the platform to change its performance characteristics will be limited by the payload capacity of the table and the maximum accelerations that are required for a particular test. These observations are consistent with those made during tests on the Athens table, and with reported practice at other shaking tables (e.g. at SUNY, Buffalo, USA) where additional static masses are often used to reduce the ratio of the specimen to platform mass and hence reduce specimen-table interaction.

The tests that measured the frequency response of the table to a constant input signal with the flexible specimen on the table showed there was little interaction between the table and the flexible specimen. The specimen itself was loaded with a mass of 3 tonnes at the top of the columns and had a total mass of 5 tonnes. No typical peak and notch resonance effect was observed in the frequency response. This was because the oil column resonance effects were swamping any other effects that were present. The only effects which the specimen had on the table response were the lowering of the resonant frequencies of the oil columns and increased damping values, as had already been observed when static mass was added to the platform, and a slight peak in the response spectrum at the natural frequency of the specimen. Figures 5.10 and 5.11 show the frequency response of the table, with the flexible specimen attached, to random noise input in only one axis at a time. On these plots the main response is the oil column resonance, and the smaller peak is the response of the specimen which is interacting with the platform response. Measurements of the platform response at several input signal amplitudes showed that the platform response (with flexible specimen attached) was not completely proportional to the input signal because of the non-linearity of the flexible specimen. The damping in the specimen increased as the shakes became larger and this effectively changed the frequency characteristics of the whole shaking table-specimen system.

The frequency response function tests brought to light several problems with the shaking table hardware itself and with the calibration procedures for the control hardware in Bristol. The aim of any calibration procedure is to achieve a flat response of the platform motion to an input signal at all frequencies. The hardware control system in Bristol did not have as many adjustments as the MTS system, and so did not have as much control over the hardware calibration as was available in Athens, ISMES or LNEC. However, as discussed in §5.5.1.2, the control software that has been developed over several years can cope with these resonance regions in the table response, and can compensate to produce the specified time history or response spectrum.

5.5.1.2 Time history response

The full series of time history test results (see Appendix A) from the testing performed in November 1993 for the Bristol table have already been published (Crewe & Taylor, 1994). All the data was plotted in a common format using the Bristol developed digital signal processing programme, DSP. Three pages of plots showing the acceleration time history, the displacement time history and the acceleration response spectrum results were produced for each axis. Each time history page (see figure 5.12 for an example) presents five graphs showing a single shaking table axis response. The top two graphs show the required and achieved platform time histories; normally these are accelerations or displacements. Beneath these two graphs is a third graph in which is an overplot of the two time histories which highlights any differences in the signals. Finally, the lowest two graphs show the amplitude and phase, respectively, of the frequency response function between the achieved and required time histories. Each response spectrum page presents four graphs. The upper two graphs show the required and achieved acceleration response spectra for 5% damping. The third graph shows an overplot of the two spectra. The bottom graph shows a simple error measure based on the percentage difference between each frequency component of the required and achieved response spectra.

The iterative time history matching process used at Bristol has already been described in detail in §4.4.2.2, but will be summarised briefly here for completeness. Depending on the frequency content of the required platform motion, the initial driving signal is taken to be the required displacement, velocity or acceleration time history. This is played through the shaking table system, ideally with the test specimen mounted, and the platform response recorded. The inverse transfer function between the platform response and drive signals is computed and multiplied by the Fourier frequency components of the required time history to produce a modified drive signal. This new drive signal is played through the shaking table and the process repeated until a satisfactory match is attained between the required and achieved motions. Two or three iterations are usually sufficient. The frequency functions are computed on 256-point, overlapped segments with Hanning windows, typically of about 2 seconds duration. This approach has the advantage that non-linear effects occurring during a transient motion can be dealt with more readily than if the frequency functions were computed over the whole time history as a single segment. It has the disadvantage that the frequency resolution is greatly reduced, meaning that lightly damped system resonances, which have narrow frequency peaks, are more difficult to control.

The results of this series of tests indicated that, with the combined hardware and software control system as it stood at that time in Bristol, it was possible to iteratively match either acceleration or displacement time histories accurately. However, if both the accelerations and displacements of the platform needed to be controlled then the fit would not be as accurate. The new hardware and software that is now beginning to be used in Bristol as this research concludes appears to have improved this situation, but there is still scope for improvements in this area of table control software.

The original software was generally able to compensate for any resonance region in the hardware control, but the matching algorithm could become unstable after several iterations, with the best results being achieved after only one or two iterations of the matching procedure. At this point both the acceleration and displacement responses were similar to the specified responses. The instability of the matching algorithms occurred when the software attempted to control table resonances at frequencies that had no components in the required signals.

The El Centro shake was only a single-axis motion, and it was iteratively matched only in one table axis at a time. Figures 5.12 and 5.13 show typical matches of acceleration time histories and response spectra for the longitudinal axes on the Bristol shaking table with no payload. The time history match is excellent. The frequency response function is also very good, being close to unity with zero phase shift, at frequencies up to 8 Hz. Above 8 Hz the frequency response is dominated by noise because of the very low energy content in the driving signal at these frequencies (§5.4.7), which leads to spurious peaks in the frequency response function. Similarly, the acceleration response spectra matches are excellent, and in this case even above 8 Hz. The response spectrum is significant since it indicates the forces, and hence responses, that a flexible specimen would experience if subjected to the platform motions. It is clear from figure 5.13 that the shaking table would not induce any significant variation in the forces on the test specimen from those required. There was no significant effect on the quality of matching when a rigid payload was mounted on the platform.

The frequency response tests described in §5.5.1.1 showed that the system resonances occur at approximately the same frequency in both the X and Y table axes. The peak system responses at the resonant frequencies are also very similar, but the general hardware gains of the two axes are dissimilar. To cope with this difference, software compensation must be used to increase the overall gain in the table Y-axis; in this way the platform will experience the same acceleration as when the signal is sent uncompensated to the X axis. This difference in hardware system gain between these two axes is probably a result of the table tuning. However, even though the internal gains in the X and Y axes are different, it is possible to iteratively match the same required response on either axis, although it is not possible to match a platform response in one axis and then use the input signal and gains from this axis in the other axis and achieve the same matched platform response.

Most of the time history testing was carried out with the three-axis Kalamata shake. Typical time history and response spectrum plots for an acceleration match of the three-axis shake, with no payload, are shown in figures 5.14 and 5.15. Once again, the time history and response spectrum matches are very good, demonstrating the ability of the Bristol control system to iteratively match multi-component shakes.

Figures 5.16, 5.17 and 5.18 show the acceleration and displacement time history matches and the response spectrum matches for a three-axis Kalamata shake with a 5 tonne flexible specimen mounted on the shaking table platform. The table was controlled to iteratively match accelerations only. The matching is good, but not as accurate as for the bare platform. The specimen had natural frequencies of 8 Hz in the longitudinal axis and 6.8 Hz in the lateral axis. Some interaction between the shaking table and specimen can be seen at these frequencies, especially in the frequency response functions and in the response spectra.

It is worth noting that the Kalamata records have little energy above 8 Hz, and therefore the driving signal would have little energy with which to control the table-specimen interaction at this frequency and above. It is not surprising, therefore, that the longitudinal response spectra show a significant peak at the natural frequency of the specimen. In the lateral axis, however, where the driving signal contains more energy around the specimen frequency of 6.8 Hz, table-specimen interaction is less pronounced, showing that the control algorithm works provided that there is sufficient energy in the initial drive signal at relevant frequencies.

The natural frequency of the shaking table platform and horizontal actuators is around 16 Hz, and this is reflected in the response spectra. Once again, the control algorithm had little energy to work with around this frequency and was therefore unable to eliminate the effects of this resonance.

The selected specimen configuration and Kalamata time history represented a severe test of the control algorithm and clearly demonstrated the latter's performance limits. This is an important finding with respect to the planning of experiments and the selection of input time histories. From a practical point of view, the Kalamata record would only be used in a real experiment where the only significant natural frequencies of the test specimen were below about 7 Hz. There would be little point in using this record if the natural frequencies of the specimen were higher.

It is believed that the response of the Bristol table could also have been improved by iteratively matching the Kalamata earthquake as a full, six-axis shake, with the rotational components matched to zero. In the test programme at Bristol, the rotational degrees of freedom were effectively uncontrolled, apart from some limited analogue cross-coupling in the hardware control system. Thus, no account was taken of the pitch and roll motions of the specimen which would have had an effect on the translational response of the platform.

This key aspect was studied further during the testing at the ISMES (see §5.5.3.1) and LNEC facilities.

The results presented above were all for tests in which the platform motions were matched to the required accelerations. The software control system at Bristol could also be used to iteratively match required displacements, but not at the same time as accelerations. Displacement matching tests were also carried out but, in general, although the displacement matching was very good, the associated acceleration responses of the platform were not as good as those obtained from the acceleration matching tests. This was because of the different frequency ranges associated with displacement and acceleration control. The former is typically dominated by frequencies up to about 5 Hz, whereas the latter is dominated by frequencies from about 10 Hz upwards. The intermediate frequency band tends to dominate velocities. Development of a new control algorithm that combined displacement, velocity and acceleration data to produce a composite control signal that encompassed the full operating frequency range of the shaking table would have been one solution to this problem. Such a system would yield much better simultaneous matching of accelerations, velocities and displacements. However, researchers at Bristol are currently working on developing a new control technique for shaking tables that is intended to overcome this problem as well as the problems of real-time control at a more fundamental hardware level (§6.7).

From the results of the series of tests it was clear that the control system at Bristol University could control either the accelerations or displacements of the shaking table platform during a three-axis shake very effectively. This issue is discussed further in §6.6. By inference, the control system could also control the platform motion in a single axis only, if this is required, for example for the El Centro shake.

Neither the additional masses placed on the platform for some of the tests nor the flexible specimen had a significant effect on the performance of the table so long as the time histories were matched under these particular loading situations. In the cases where matched time histories were run through the table with a changed payload condition, the platform response was not the same as previously recorded. This change was caused by the shift in the resonant frequencies of the system, and a shake that had been matched to one particular system response would not work if the overall system characteristics were changed.

5.5.1.3 Response spectrum fidelity

Much of the commercial work carried out on the Bristol shaking table requires that an arbitrary acceleration time history be produced that conforms to a specified response spectrum. The computer programme THS was developed, in Bristol, over several years to generate these time histories, and in general version 2.01 of this software will generate a platform motion to meet these requirements to within 2% or 3%. This software was upgraded during this research, so that it has been possible to iteratively match some very severe shakes that Bristol has been asked to perform. This software will also compensate for the table control system resonances, although it cannot control the response of the platform so effectively at the harmonics of these frequencies. The maximum achievable values for a response spectrum are limited only by the maximum achievable displacements, velocities and accelerations in the horizontal and vertical axes. Additional research into the ability of the Bristol shaking table to meet the requirements of response spectra testing was not felt to be necessary as part of this research programme, as many previous tests at Bristol have shown how effective this software can be.

5.5.1.4 Summary of system characteristics and evaluation of system performance

The general performance of the Bristol shaking table in producing specified time histories and response spectra was very good at the time these tests were performed. This indicated that the software controlling the system was effective, and that effort to improve the table performance would best be spent on improving the hardware control system. By improving the hardware control it should also be possible to improve the overall performance and stability of the control software. Based on the results of this research, the upgrading of the shaking table control hardware has now taken place with the analogue controller being replaced by a much more adaptable digital controller. This controller was specifically designed to allow the testing and development of new active control techniques for shaking tables. Another recent modification to the shaking table at Bristol is the addition of differential pressure gauges across the actuators which have significantly improved the ease with which the table can be tuned.

During this series of tests there were no problems with the stability of the analogue control hardware as Clark and Cross (1984) suggested might be the case. Nor have problems been encountered with the new digital controller, although further tests at the extremes of the table's performance specification may be needed in the future to confirm that the new hardware is stable at all frequencies. However, it is not anticipated that there would be any problems in this respect, as other tests over the years have not revealed this type of problem, any instabilities in the table control system having been caused by problems with the software control system.

5.5.2 Athens site

5.5.2.1 Frequency Response

The full series of frequency response test results for the Athens table have already been published (Crewe & Taylor, 1994). Over 80 sets of measurements were taken in December 1993 to explore the frequency response of the Athens shaking table when carrying a variety of rigid and flexible payloads (see Appendix A).

As discussed previously (§4.4.2.1), the analogue hardware control system of the Athens shaking table allows subtle control of the frequency response of the table. It is possible to control independently the gains and systems delays (or phase response) associated with displacement, velocity and acceleration feedbacks. In effect this gives the operator the ability to adjust the frequency response in three frequency bandwidths; low frequencies (dominated by displacements), intermediate frequencies (dominated by velocities) and higher frequencies (dominated by accelerations). The effectiveness of this tuning was demonstrated by the frequency response plots for the bare platform. Figure 5.19 shows a typical transfer system function for the bare platform that was obtained after tuning the system. Table 5.11 summarises the results of all the transfer functions measured for the bare platform, including the checks made for cross-coupling of axis motions. The values given show the maximum and minimum values of the transfer functions obtained over the operating range of the table (up to 25 Hz). Similar results for rigid payloads are shown in table 5.12. With careful adjustment of the analogue controls it was possible to produce a very flat frequency response over the whole operating frequency range of the table, although this became progressively harder to achieve as the payload on the platform

Table 5.11 Ranges of amplification factors for single-axis random excitation of the Athens table with various payloads after tuning

Excitation	**Response Axis**											
Axis	**X (Longitudinal)**		**Y (Lateral)**		**Z (Vertical)**		**R (Roll)**		**P (Pitch)**		**W (Yaw)**	
	Min Amplification	Max Amplification	Min Amplification	Max Amplification	Min Amplification	Max Amplification	Min Amplification	Max Amplification	Min Amplification	Max Amplification	Min Amplification	Max Amplificatio n
X (Longitudinal)	**0.88**	**1.15**	0.00	0.03	0.00	0.03	0.00	0.02	0.00	0.03	0.00	0.08
Y (Lateral)	0.00	0.03	**0.83**	**1.20**	0.00	0.02	0.00	0.03	0.00	0.01	0.00	0.12
Z (Vertical)	0.00	0 06	0.00	0.06	**0.88**	**1.12**	0.00	0.15	0.00	0.12	0.00	0.053
R (Roll)			0.06	0.60	0.00	0.08	**0.86**	**1.23**				
P (Pitch)	0.06	0.11			0.00	0.68			**0.78**	**0.91**		
W (Yaw)	0.01	0.11	0.00	0.21							**0.86**	**1.20**

Table 5.12 Ranges of amplification factors for single-axis random excitation of the Athens table with various payloads after tuning

Excitation and Response Axis	Payload Bare platform		4 tonnes		8 tonnes	
	Min Amplification	Max Amplification	Min Amplificatio n	Max Amplification	Min Amplification	Max Amplification
X (Longitudinal)	**0.88**	**1.15**	**0.72**	**1.40**	**0.42**	**1.20**
Y (Lateral)	**0.83**	**1.20**	**0.43**	**1.21**	**0.48**	**1.23**
Z (Vertical)	**0.88**	**1.12**	**0.81**	**1.00**	**0.52**	**1.20**
P (Pitch)	**0.78**	**0.91**	**0.92**	**1.42**	**0.92**	**2.65**
R (Roll)	**0.86**	**1.23**	**0.88**	**1.61**	**0.89**	**2.34**
W (Yaw)	**0.83**	**1.20**	**0.94**	**1.18**	**0.63**	**1.37**

increased. The Athens table did not suffer from effect of the oil column resonances (§4.3.2.6) that were present in the Bristol table (§5.5.1.1) as they could be compensated for in the various feedback loops.

With rigid payloads, cross-coupling of the various axes was negligible. With flexible payloads, the frequency response was still good, but a classical 'peak and notch' response was seen around the natural frequency of the test specimen (9.187 Hz and 7.25 Hz in the major and minor axes respectively). At these frequencies the specimen tended to drive the shaking table and giving rise to a 'peak' in the frequency response function followed immediately by a trough or 'notch' (figure 5.20). Once again, by careful tuning of the analogue control hardware it was possible to minimise this effect, although it was not possible to eliminate the effect entirely (figure 5.21). Cross-coupling of motions between axes was more noticeable with flexible payloads, particularly for the rotational axes with respect to their associated translational axes. Again, careful tuning of the analogue control system minimised this problem.

From an operational viewpoint, therefore, the analogue control hardware of the Athens table should be tuned, with the test specimen mounted, to optimise the frequency response functions in the various axes and to minimise cross-coupling effects. This can be done fairly rapidly using low level random noise inputs and an on-line Fourier analyser. Once the analogue system is properly tuned, the seismic control software is better able to iteratively match required time histories.

5.5.2.2 Time history response

The full series of time history test results (see Appendix A) from the testing performed in December 1993 for the Athens table have already been published (Crewe & Taylor, 1994).

The seismic control software of the Athens table works in a similar manner to the Bristol software. The inverse transfer function of the whole shaking table-specimen system is measured and used to modify the driving signal in an iterative manner until a satisfactory match is obtained between the required and achieved platform motions. The main difference between the Athens and Bristol approaches is that the Athens software computes the frequency responses from the whole time history, whereas the Bristol system works on overlapped segments of the time history.

Figures 5.22, 5.23 and 5.24 show the acceleration, displacement and response spectrum matches, respectively, for the El Centro record applied in the lateral axis with zero payload. The table was controlled to iteratively match the acceleration time history. The acceleration match is excellent, although the achieved acceleration response is slightly polluted by electrical noise on the analogue signal. The frequency response function is almost unity from 1 Hz to 8 Hz, and thereafter is dominated by numerical noise due to the low signal energies above 8 Hz. Unlike the Bristol response functions, the phase of the Athens response function is not zero between 1 Hz and 8 Hz. It is approximately linear, which might be due to a slight sampling delay between the output D/A converters and input A/D converters on the control computer. Such a phase shift is unimportant with respect to actual shaking table experiments. The displacement match is good, but not as close as the acceleration match. This is because the control software was run in acceleration control mode. As was the case for the Bristol table, it was found that it was possible to produce excellent matches for either accelerations or displacements, but not always for both simultaneously. The response spectrum plots show excellent agreement between the required and achieved spectra.

Figures 5.25, 5.26 and 5.27 show the acceleration and displacement time histories and response spectrum matches for one axis of the three-axis Kalamata record with a bare platform. Once again, the matches are excellent, including in this case the displacement records. The good displacement match, even though the table was controlled in acceleration mode, is probably because the Kalamata record does not have such strong low frequency components as the El Centro record.

Figures 5.28, 5.29 and 5.30 show the acceleration and displacement time histories and response spectrum matches for one axis of the three-axis Kalamata record with a 5 tonne flexible specimen on the platform. As can be seen the desired and achieved platform motions are almost identical both in the time and frequency domain. The efficiency of the analogue tuning of the Athens table is reflected in the response spectrum plot (figure 5.30) which shows little deviation around the natural frequencies of the specimen (6.8 Hz and 8 Hz) and is not affected by any system resonances. This confirms observations made by Takahashi et al. (1974) that although the transfer function of a shaking table system may show significant table-specimen interaction this does not necessarily have a significant effect on the ability of a shaking table to reproduce specific time histories.

Overall, throughout the extensive series of seismic tests, it was found that the Athens shaking table produced very good matches between required and achieved time histories with only small errors in either time or frequency domains.

5.5.2.3 Response spectrum fidelity

The procedure used at Athens to iteratively match required response spectra is based on the same technique as is used to match the time histories. At Athens, an artificial acceleration time history is generated numerically to envelop the required response spectrum. This time history is then matched on the shaking table. As has already been shown, the Athens iterative time history matching leads to excellent enveloping of the associated response spectra. Therefore, provided the artificial, spectrum-compatible time history adequately envelops the required response spectrum, the achieved test response spectrum should be satisfactory.

5.5.2.4 Summary of system characteristics and evaluation of system performance

The Athens tests demonstrated the excellent performance that can be achieved from the MTS analogue and software control systems, provided that the table operator understands the characteristics of the system. A major drawback of the system, however, was the low speed of the PDP11/34 minicomputer. The necessary computations to iteratively match time histories could take up to one hour, considerably reducing the productivity of this shaking table. In contrast, the similar computations carried out on the other shaking tables take only a few tens of seconds on 486 66 MHz processor PCs. Following this research, steps have been taken to replace the PDP11/34 on the grounds of speed and maintenance costs. While this was not a simple task, because the MTS software is proprietary and the source code was not available to the Athens laboratory, software developed in the other three laboratories has now been successfully implemented in Athens.

5.5.3 ISMES site

5.5.3.1 Frequency Response

The full series of frequency response test results for the ISMES table have already been published and can be found in the "Interlaboratory Standardisation tests on the 6 DOFs Shaking table MASTER" (ISMES, 1996). Almost 80 sets of measurements were taken in November 1995 to explore the frequency response of the ISMES shaking table when carrying a variety of rigid and flexible payloads (see Appendix A).

Table 5.13 lists the frequencies and relevant amplitudes of the platform response for the three configurations (bare platform, 8 tonne rigid payload and 8 tonne flexible specimen) evaluated after tuning of the table had taken place. The frequency transfer functions are listed for each axis when excited individually by a random signal. A typical frequency transfer function is shown in figure 5.31.

The dynamic characteristics of the flexible specimen (table 5.14) were obtained by mounting several accelerometers on the specimen. The resonant frequencies of the 8 tonne flexible specimen in its major and minor axes were 7.6 Hz and 6.4 Hz.

Table 5.13 Natural frequencies of the MASTER shaking table

	Natural frequencies and amplifications in the MASTER shaking table					
Excitation axis	**Bare Platform**		**Rigid Payload**		**Flexible Payload**	
	Frequency (Hz)	Amplification	Frequency (Hz)	Amplification	Frequency (Hz)	Amplification
X (Long.)	20-35	1.3	19	1.7	22	2.8
Y (Lateral)	26	1.6	20	2.1	24	2.8
Z (Vertical)	36	1.7	22	1.8	22 to 58.5	2.6 to 6
R (Roll)	40	1.6	36	1.4	40	2.4
P (Pitch)	51	1.8	40	1.7	45	2.6
W (Yaw)	43	1.8	30	1.4	40	3.1

Table 5.14 Dynamic characteristics of the flexible specimen as used in ISMES

Excitation	**1st Mode**		**2nd Mode**	
	Frequency (Hz)	**Damping ξ (%)**	**Frequency (Hz)**	**Damping ξ (%)**
Major axis	7.6	3.7	31.5	0.7
Minor axis	6.4	16.0 *	47.6	1.0

* The high damping in this axis was caused by slippage and the associated friction in the bolted joints in the specimen.

The MASTER table analogue console, made by MTS, allows fine adjustment of all the gains in the feedback control loops. In the low frequency range the control is dominated by the displacement feedback, at intermediate frequencies control is dominated by the velocity feedback and at high frequencies control is dominated by the acceleration feedback. However, it was evident that, although many adjustments could be made, a unity gain for the excited axis could not be achieved and the effects of the table resonances could not be completely cancelled. Furthermore, the values of the amplifications increased as the conditions changed from the bare platform to the flexible payload condition. This was probably due to the fact that, in order to reduce the effects of the specimen resonances, the tuning was optimised for these frequencies, resulting in a poorer calibration at the other table frequencies. Tuning the ISMES table was therefore a compromise operation covering the whole frequency range of the table. The result of this was that the maximum amplification always occurred at the resonant frequencies of the table, even though the specimen resonances were very marked. Ultimately, amplification factors which did not exceed the values of 2 or 3 were considered to be a good compromise, for this table. In any case, the digital control system at ISMES could easily compensate for these levels of amplification in the system.

Tuning of the ISMES table was very dependent on the experience of the operator and on the conditions in which the table had been maintained. A comparison between figure 5.32 and figure 5.33 shows the large difference in tuning that results from the addition of a flexible payload on the platform. It can be seen that because of inadequacy of the pitch control circuitry high amplifications result in poor coherences; in this case the compensation processes used by the digital control system may diverge, with the result that an accurate reproduction of the required platform motion cannot be achieved.

In the tests with the flexible payload, the frequency transfer functions between the axes being excited and the other table axes also show several anomalous peaks. Ideally these transfer functions should have a value of zero. This cross-coupling between axes occurred regardless of which table axis was excited, and although accurate tuning of the system reduced these effects it could not remove them completely. In general it was possible to control the motions in the coupled axes down to about 20% of the maximum amplification of the excited axis. In most cases proper tuning also limited their influence at the higher frequency ranges, which are beyond the normal seismic frequency range.

The frequency transfer functions of the vertical axis when excited with random noise at three different amplitudes (0.1, 0.3 and 0.4 m/s^2) under bare platform conditions, showed slight variations which could be attributed to the non-linear behaviour of the electro-mechanical system. This suggests that it might be desirable to adjust the table tuning during a series of tests where the amplitude gradually increases. The results obtained in Bristol and Athens do not show such significant non-linear behaviour except towards the extremes of the table performance ranges, so that table re-tuning is not necessary on these tables unless there is a significant increase in the desired amplitude of the shake. The addition of a specimen whose response changes from being linear to non-linear as the amplitude of the platform motion increases will, however, affect the overall behaviour of the table, which will require significant re-tuning if its performance is not to be adversely affected.

5.5.3.2 Time history response

The full series of time history test results (see Appendix A) from the testing performed in November 1995 for the ISMES table have already been published (ISMES, 1996). During these tests measurements were taken to explore the response of the table under a variety of load conditions, and a summary of the performance of the table during these tests in given in table 5.15.

When the table was controlled in acceleration mode, the errors in the reproduction of the required acceleration time histories were very small (see figure 5.34 for a typical example), and in the worst case the error was only 12% of the peak acceleration. The reproduction of the displacement time histories is also very good; and in fact, except for the results of the tests for the bare platform and rigid payload, where an error was made in the choice of frequency range for the generated drives, the obtained displacements were within 14% of the peak required displacement. When the table was controlled in displacement mode, the errors of the obtained displacements were as low as the 3% of the maximum displacement (see figure 5.35). In this case, however, higher acceleration errors were recorded (up to 25% of the peak acceleration).

Compensation of the motions caused by cross-coupling of the tables axes (made by controlling all 6 axes) resulted, in general, in a lower accuracy in the reproduction of the main X, Y and Z time histories. This might be due to less accurate computation of the system transfer functions, suggesting the need for a higher number of averages to be taken during the pre-tests.

The accuracy of the platform motions obtained was mainly due to the compensation methods performed by the digital control system. In fact, when no compensations and corrections were used, as in the test Kala8t0 (see figure 5.36), the error in platform acceleration was much higher (more than 100%), indicating that even with good tuning of the table additional corrections are required to produce adequate control. With software

Table 5.15 Summary of the performance of the ISMES table during the time history tests

Test	**Maximum time history error (%) (1)**						**Maximum spectrum error (%) (2)**			**RMS acceleration of undesired motions (m/s^2)**		
	X		**Y**		**Z**							
	acc.	displ.	acc.	displ.	acc.	displ.	X	Y	Z	Yaw	Pitch	Roll
Kalamata	8	57	6	31	6	75	12 (20)	13 (32)	25 (28)	0.019	0.045	0.029
Kala 8t0	60	71	50	45	112	100	57 (17)	40 (14)	210 (23)	0.019	0.040	0.033
Kala 8t3	8	54	7	31	12	67	11 (17)	15 (14)	6 (12)	0.022	0.034	0.025
Kala 8t6	4	21	7	6	11	23	7 (20)	9 (14)	11 (20)	0.013	0.039	0.028
Kala 8t6d	10	1	20	3	25	3	18 (18)	45 (18)	21 (19)	0.044	0.037	0.031
Flex 3	7	4	6	5	6	10	13 (18)	10 (23)	8 (28)	0.017	0.066	0.064
Flex 6d	10	10	9	1	26	2	13 (28)	32 (22)	18 (27)	0.031	0.082	0.049
Flex 6	12	12	11	8	11	14	11 (18)	14 (22)	13 (18)	0.015	0.037	0.060

(1) Referred to the peak amplitude of the time history

(2) The number in brackets shows the frequency (Hz) component with the maximum error

compensation the frequency transfer functions between achieved and required accelerations (figure 5.37) show zero phase and unit (1) amplitude up to about 15 Hz, beyond the frequency range of the reference spectra, indicating that the frequency distortions of the whole excitation system are well compensated.

The time history spectra calculated from the tests show similar characteristics to the results of the acceleration matching tests. When the table was controlled in acceleration mode, the spectral matching is very good (see figure 5.38). The highest errors (25% in the worst case) occurred at a frequency above that containing any significant excitation energy. Higher errors (max. 45%) were found when the table was controlled in displacement mode (see figure 5.39). When no digital corrections or compensations were used the error became as high as 210% in the vertical axis (see figure 5.40). No significant differences in the spectra matching were noticed between three-axis and six-axis control of the table (see figures 5.38 and 5.41).

Based on the results of the tests performed in Bristol and Athens, for the series of tests performed in ISMES additional measurements were made of the motions in the three rotational axes. These measurements were used to look specifically at how accurately the table control systems could eliminate the undesired motions in these axes. The accelerations recorded in the rotational axes were almost at electrical noise amplitudes in the case of tests with bare platform or with rigid payload; but some increase in the level of accelerations was found with the flexible payload on the platform, especially in the pitch and roll axes. However, even in this case, the peak values of the undesired motion with respect to the corresponding translation was less than 15%.

The tests performed showed that the presence of undesired rotational motions has a very strong influence on the response of the specimen. It was found that the frequency content of the pitch motion during control of only the translational degrees of freedom showed the almost exclusive presence of a component at the resonance frequency (7.6 Hz) of the specimen in longitudinal direction (see figure 5.42). This component was significantly lower when all six axes were controlled (see figure 5.43). When considering the frequency transfer functions of the responses measured at the top of the specimen in the two main directions X and Y, it can be concluded that the resonance frequencies of the specimen are about 10% lower when three axes are controlled (figure 5.44), and the corresponding amplification of motion was 4 to 5 times lower than when all six axes were controlled (figure 5.45). This is clearly caused by table-specimen interaction, with the resulting absorption of energy (additional damping) into the undesired motions.

The problems of the interaction between a specimen and a shaking table are quite well known and have been studied analytically by a several researchers (see Chapter 2) but there has been little experimental work on how significant this problem actually is with regard to real shaking table tests. The tests performed at ISMES showed that even a relatively small specimen (72% of the mass of the platform and 27% of the table capacity) caused significant table-specimen interaction, enough to have seriously affected the response of the model. The influence of the undesired rotational motions can therefore, in some cases, affect the reliability of the shaking table tests; so these motions should be controlled using the matching software (§6.4) or at least monitored so that they can be taken into account in any subsequent analysis of the experimental data (§6.5). In the past, a single-axis excitation was accepted as sufficiently representative of an earthquake phenomenon. In recent years more severe requirements, due to a scientific need for a more realistic excitation, necessitated the multi-axis (translational) reproduction of the earthquakes. This has become possible because modern technology has enabled construction of shaking tables capable of moving along each of their six degrees of freedom. Consequently, the problem of controlling all the degrees of freedom has become very important, especially with reference to the cross-coupling between translations and rotations. The tests performed at ISMES confirmed the necessity of controlling all the degrees of freedom of the table in order to minimise the effects of table-specimen interaction.

5.5.3.3 Summary of system characteristics and evaluation of system performance

The ISMES tests demonstrated the excellent performance that could be achieved from the MTS analogue hardware and NewMACS software control systems, provided that the table operator understands the characteristics of the system. The testing in ISMES also confirmed that it is very important to correctly tune the shaking table in order to obtain acceptable results. It also highlighted some possible limitations of shaking table tests, whose effectiveness depends on the control of non-linearity in the behaviour of table and specimen. It has to be emphasised that, in order to avoid damaging the specimen before testing starts, the tuning must be performed at a reduced amplitude with respect to the final testing amplitude. Corrections of the calibrations may then be necessary when the testing amplitude is progressively increased. Furthermore, the effectiveness of the tuning on the accuracy of reproduction of specified time histories was reduced when the specimen had a strong interaction with the table. In this case, unless a digital control system able to control the undesired motions of the platform is available, the results obtained during testing can be seriously affected. Finally, the tuning procedure can only be seen as a compromise, to obtain a generally acceptable behaviour of the table over the whole frequency range of interest.

5.5.4 LNEC site

The tests performed at LNEC were somewhat more detailed than those performed at the other three sites as they were the first tests to be performed on the LNEC table after commissioning. Also, as the LNEC table is a three-axis table with a torque tube system restraining the rotational platform motions, it was of particularly interest to see how effectively these rotational degrees of freedom were restrained. The operators at LNEC were also developing a finite element (FE) model of their new shaking table as this phase of the test programme commenced. In order to calibrate their model it was particularly important to take additional measurements of the frequency responses of the various parts of the shaking table so that the various elements in the FE model could be modelled correctly. The FE model produced and calibrated using the results of this research has subsequently been used to check the effect of test specimens on table performance before testing on this table commences.

5.5.4.1 Frequency response

The full series of frequency response test results for the LNEC table have already been published and can be found in the "Characterisation of the new LNEC shaking table" (LNEC, 1996). Many sets of measurements were taken in February 1996 to explore the frequency response of the bare LNEC shaking table and the table when carrying a flexible payload (see Appendix A).

The evaluation of the frequency response functions (FRFs) at LNEC were carried out using in-house software which compared the response of the shaking table in each of the six degrees of freedom with the single-axis input signal. Key results from the analysis of the frequency response function tests over the operating frequency range of the LNEC table (0-20 Hz) are given below.

Analysis of the FRFs of the three main table axes showed that there were three translation natural frequencies of the system under analysis, close to 11 Hz, 13.5 Hz and 8.5 Hz for the longitudinal, lateral and vertical axes respectively. In the lateral direction, there was also some anti-resonance at a frequency of 11 Hz.

For the excitation in the X (Longitudinal) axis, the following key characteristics were observed.

- Cross-coupling between the X (Longitudinal) axis and P (Pitch) axis was evident over the frequency range, 10 Hz to 22 Hz. The maximum resonance had a gain of 6 rads/s^2 per g occurring at 18 Hz.
- No evidence of cross-coupling was found between X (Longitudinal) axis and R (Roll) axis.
- Cross-coupling between X (Longitudinal) axis and W (Yaw) axis showed a clear peak value of 0.8 rads/sec^2 per g at 11 Hz, followed immediately by anti-resonance at 13 Hz.

For the excitation in the Y (Lateral) axis, the following key characteristics were observed:

- A small amount of cross-coupling between Y (Lateral) axis and Z (Vertical) axis was noticed, with gains of between 0.02 and 0.08 in the frequency band of 16-18 Hz, where the frequency response reached its highest value.
- Cross-coupling between Y (Lateral) axis and X (Longitudinal) axis was apparent, with peak resonance values of 0.3 and 0.09 at frequencies of 11 Hz and 15.5 Hz respectively.
- The cross-coupling between Y (Lateral) axis and P (Pitch) axis was about 0.1 rads/s^2 per g throughout the frequency range.
- The cross-coupling between Y (Lateral) axis and R (Roll) axis showed a peak value of 6 rads/s^2 per g at 18 Hz. The damping of this resonance was relatively high (estimated at 9%).

- The cross-coupling between Y (Lateral) axis and W (Yaw) axis showed a clear peak value of 7 rads/s^2 per g at 11 Hz. Damping at this frequency was also relatively high, (estimated at 7% of critical).

For the excitation in the Z (Vertical) axis, the following key characteristics were observed:

- Cross-coupling between Z (Vertical) axis and X (Longitudinal) axis showed a gain of 0.02 in the frequency range 0-10 Hz. Thereafter, the coupling increased up to a value of 0.3 at 21 Hz.
- Cross-coupling between Z (Vertical) axis and P (Pitch) axis showed resonance (2 rads/s^2 per g) and anti-resonance (0.1 rads/s^2 per g) at frequencies of 21 Hz and 8.5 Hz respectively.
- Cross-coupling between Z (Vertical) axis and R (Roll) axis was relatively low (0.2 rads/s^2 per g).
- Cross-coupling between Z (Vertical) axis and W (Yaw) axis was also relatively low, about 0.1 rads/s^2 per g in the frequency range of interest.

Table 5.16 provides a summary of the resonances and gains listed above.

Table 5.16 Resonance frequencies and amplification factors for single-axis pink noise excitation before tuning.

	Excitation Direction					
Axis	**X (Longitudinal)**		**Y (Lateral)**		**Z (Vertical)**	
	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification	Freq. (Hz)	Amplification
X (Longitudinal)	**11 Hz**	**90% g/mm**	11 Hz	0.3 g/g	21 Hz	0.3 g/g
Y (Lateral)			**13.5 Hz**	**22.5% g/mm**		
Z (Vertical)				negligible	**8.5 Hz**	**22.5% g/mm**
R (Roll)		negligible	18 Hz	6 rads/s^2 /g		negligible
P (Pitch)	18 Hz	6 rads/s^2 /g		negligible	21 Hz	10 rads/s^2 /g
W (Yaw)	11 Hz	0.8 rads/s^2 /g	11 Hz	7 rads/s^2 /g		negligible

The diagonal terms highlighted in bold correspond to the transfer functions of the main table axes with other values showing the extent of the cross-coupling between axes. Although no reference has been made to the upper off-diagonal terms of the frequency response matrix in table 5.16 (i.e. lateral / longitudinal, lateral / vertical, vertical / longitudinal), these relationships were briefly analysed and showed, as expected, anti-resonances in the frequency response where their symmetrical terms exhibit resonance.

It can be seen that the three main translational axes have significantly different calibrations, with the longitudinal axis having a response four times higher that the other two axes. This is caused by the much higher capacity of this actuator and means that it is impossible to interchange drive signals between different axes on this table without some additional software which can control the overall system gains. There is also some interaction between the translational and rotational table axes. Since the rotational components in this table are controlled entirely passively this cross-coupling cannot be avoided without making alterations to the torque tube system. However, the cross-coupling was not particularly large and would not be expected to increase significantly even if a specimen that would interact significantly with an equivalent six-axis table were tested on this table.

5.5.4.2 Time history response

The full series of time history test results (see Appendix A) from the testing performed in February 1996 for the LNEC table have already been published and can be found in the "Characterisation of the new LNEC shaking table" (LNEC, 1996).

The time history tests determined the ability of the earthquake simulator to reproduce accurately the target motions in the three translational directions (longitudinal, lateral and vertical). Specifically, these tests looked at the ability of the software control system to cope with the resonances and anti-resonances found during the evaluation of the frequency response functions, discussed in the previous section (§5.5.4.1). The software control system only attempted to flatten the global frequency response function and improve the coherence between target and achieved motions for the three actively-controlled DOFs (longitudinal, lateral and vertical). It was expected that the software control system would have only a minor influence on the passively-controlled DOFs (pitch, roll and yaw). For this reason, few references to the cross-coupling involving the platform rotations have been made in this section.

In order to improve the first attempt at the reproduction of a defined shake on the LNEC table, the transfer functions of the three translational axes were measured, inverted and then used to pre-adjust the drive signal before testing. Theoretically this technique would compensate for the errors in the global system FRF and would produce an exact reproduction of the three-axis Kalamata earthquake and a single-axis white noise test on the shaking table. The white noise test was an additional test performed at this site to check the performance of the new software control system in iteratively matching a signal that had significant amplitude at all the frequencies within the operating range of this table. All of the time history reproduction tests were performed on the bare shaking table platform and also with the 8 tonne flexible specimen. Typical results, in terms of response spectra achieved, are shown in figures 5.46 and 5.47 for a single-axis test of the reproduction of random white noise acceleration and in figures 5.48 and 5.49 for a three-axis test of the reproduction of the Kalamata earthquake.

It should be stressed that for single random white noise tests the adaptive software control procedures were carried out for each translation axis independently, and that the other translational axes were only controlled by the hardware system. No zero input target signals were imposed on the other axes in order to check whether the platform motions could be effectively controlled without any additional software control. The performance of the software control system in controlling all the three translational axes simultaneously was only considered in tests carried out with the Kalamata earthquake.

The following key characteristics were observed for the single-axis white noise tests:

1) A significant reduction in the amplitude of the system resonances occurred in all the three axes in the frequency range of target motions, especially in the lateral direction when software compensation was used to improve the platform response.

2) However, in the vertical direction, some amplification ($\approx$1.7) at 8.5 Hz remained, and a resonance frequency of 16.7 Hz with a damping ratio of 10% was clearly evident.

3) It was found that the system amplification for the single-axis tests in the longitudinal direction was lower than that found in the frequency response tests. The resonance at

approximately 16.7 Hz was the same in the two tests, although it was associated with a higher value of the damping ratio (≈20%) in the time history tests. This higher damping was probably caused by an increase in friction associated with the larger displacements of the platform.

4) For the tests without the specimen, no significant change occurred in the magnitude, phase or coherence of the cross-coupling between translational and rotational axes, either before or after any software adaptation took place. This was as expected as the rotational axes were only controlled passively.

5) The presence of the specimen apparently had only a minimal effect on the controlled axes. However, in the other axes the effect was more pronounced at the resonant frequencies of the specimen (5.76 Hz in the lateral direction and 7.13 Hz in the longitudinal).

For the three-axis tests conducted with the Kalamata earthquake (e.g. figures 5.50, 5.51 and 5.52), it was found that:

1) The matching between the target and achieved motions was not as good as was achieved in the single-axis tests. In fact, a more complicated system of resonances and cross-coupling between axes was apparent in the three-axis tests.

2) The matching was best in the range of the constant velocity of the response spectra (between 0.5 Hz and 5 Hz) where most of the energy of Kalamata earthquake is concentrated.

3) For frequencies below 0.5 Hz and frequencies above 5 Hz, the achieved response spectra were higher than the respective targets, with an amplification factor of about 2.0.

4) The peak values of pitch, roll and yaw for the Kalamata three-axis test without the specimen were 0.43, 0.28 and 0.39 rads/s^2, respectively. The peak values of pitch, roll and yaw for the Kalamata three-axis test with the specimen were 1.23, 0.52 and 0.39 rads/s^2, respectively. This indicates that cross-coupling between the X (Longitudinal) axis and the P (Pitch) axis, and between the Y (Lateral) axis and the R (Roll) axis, was occurring and was not completely stopped by the torque tube restraints. The influence of the specimen was also increasing the pitching, rolling and yawing of the table platform as more loading had to be carried by the torque tube system.

5.5.4.3 Summary of system characteristics and evaluation of system performance

From the tests performed on the LNEC table it was possible to analyse the main characteristics of the newly built LNEC shaking table.

The natural frequencies of the various axes of the shaking tables obtained from the test result, and the six first natural frequencies obtained from a finite element (FE) model (LNEC, 1996) of the shaking table (figure 5.53), are shown in table 5.17. A typical mode shape, generated by the FE analysis, is shown in figure 5.54. The values in table 5.17 clearly show that there were some significant errors in the original FE model that was created based on the design specification of the shaking table. It should be noted that this FE analysis was performed before the table was built and as such was used in the development of the table design.

Table 5.17 The first six measured and computed natural frequencies of the LNEC shaking table.

Axis	**Measured** f_m **(Hz)**	**Computed** f_c **(Hz)**	**Error** $(f_c - f_m)/f_m$
X (Longitudinal)	11	12.1	10%
Y (Lateral)	13.5	10.3	-24%
Z (Vertical)	8.5	7.0	-18%
R (Roll)	18	27.2	51%
P (Pitch)	18	27.6	51%
W (Yaw)	11	15.1	37%

As a result of the shaking table tests, in the lateral and vertical directions, the stiffness of the bar elements used in the FE analysis to model the oil columns in the actuators was increased, while in the longitudinal direction that actuator rigidity was decreased. An increase in the stiffness of the lateral and vertical actuators could be caused by better performance of the hardware control system than was initially assumed in the FE model. On the other hand, in the longitudinal direction, a different actuator (1000 kN) and servovalve (three stages) to that originally used in the FE model could explain a decrease in that stiffness (in other words, the hardware control system did not perform as well as expected).

Looking at the three natural modes of vibration in which platform rotations predominate, either the rotation-restraining systems modelled (i.e. torsion bars, cranks, connecting rods and bearings) were stiffer than those in the table, or the FE model had a lower rotational inertia than the table. This would explain the higher frequencies that were obtained from the FE model.

It was observed that the rotational resonances were all associated with relatively high damping values (>7% critical) as compared to what might be expected (≈5%) for a structural steel system subjected to earthquake loading. It was also noted that higher damping occurred at the roll and pitch natural frequencies than at the yaw natural frequency, indicating that these effects were likely to be related to the bearings at the end of the long rods. It was therefore assumed that a portion of the dead load of the platform imposed higher friction in the bearings for the yaw component than for the pitch and roll.

The influence of these resonance frequencies on the peak values of platform rotations was estimated by a more detailed analysis of the results from Kalamata three-axis tests, with and without the influence of the overturning moments produced by the test specimen. Rotational components of the platform centre were used to assess the induced displacement differences at the platform edges (roughly at ±2.5 m from the centre). Pitch and roll were expressed as differences in vertical-edge displacements, and the yaw as the differences in horizontal motions at the same points.

In this way, the rotations detailed in point 4 of §5.5.4.2, which summarises the Kalamata test results, produce the positive and negative platform-edge displacements shown in the first two rows of table 5.18. The table also shows the percentages of those values, related to the peak displacements of Kalamata earthquake in the respective direction (vertical, for pitch and roll and, longitudinal and lateral for yaw).

Looking at the values in table 5.18 the following points deserve attention. It is evident that the peak rotations are not particularly high, although it remains to be seen whether these

Table 5.18 Analysis of results in terms of the measured platform rotational components; three-axis tests, conducted for Kalamata earthquake with and without model (PGD = peak ground displacement)

	R (Roll)	**P (Pitch)**	**W (Yaw)**
Bare platform	0.5 mm	< 0.05 mm	0.75 mm
Table with 8 tonne specimen	1.0 mm	1.5 mm	0.75 mm
Kalamata signal PGD	12 mm (Z)	12 mm (Z)	67 mm (X); 43 mm (Y)
Bare platform - % of Kalamata PGD	4%	< 0.4%	1% (X); 2% (Y)
Table with 8 tonne specimen - % of Kalamata PGD	8%	13%	1% (X); 2% (Y)

levels of rotations are significant with regard to the behaviour of specimens being tested on the table. It should be noted, however, that the estimated overturning moments during the Kalamata earthquake test, induced by the presence of the model, were less than 140 kNm, and therefore were not very large. The flexible specimen used for these tests was not really large enough to provide a significantly severe test of this table (table 5.7). However, by using the same specimen as was used at the other sites comparison of results was easier. If a more massive test specimen were tested then it is reasonable to assume that more significant rotations would be seen on this table. In addition, if a specimen with a mass closer to the capacity of the platform were tested then natural frequencies of rotational axes of the platform, which were quite low with no payload on the platform (11 Hz and 18 Hz - see table 5.17), would be even lower (as was seen at ISMES - table 5.13). If the lower rotational natural frequencies of the shaking table were then to correspond with any of the natural frequencies of the larger specimen much higher table-specimen interaction would be expected which would increase the rotational components of the LNEC table.

By analysing the results of the time history tests it was possible to quantify, in more general terms, the fidelity of the overall control system used at the LNEC facility. Tables 5.19 and 5.20 give two different measures of the accuracy of the iterative matching process at LNEC for the White noise and Kalamata earthquake tests. Table 5.19 gives the mean values of the ratio of the achieved to target velocity spectra calculated at 74 evenly spaced intervals between 0.05 Hz and 50 Hz These coefficients measure the linear relationship between the desired and achieved spectra. Since these values do not quantify any temporal errors occurring between the desired and achieved motions throughout the signal reproduction on the platform another coefficient was also used to define the fidelity of the control system at LNEC. Table 5.20 shows the average of the ratios of the mean squared values of achieved and target displacements at every time interval during the tests.

Table 5.19 Summary of the correlations between achieved and target velocity response spectra ordinates.

Signal	**Single-axis White Noise Signal**			**Three-axis Kalamata Signal**
Axis	**Lateral**	**Vertical**	**Longitudinal**	**Three-axis shake**
Lat. without model	1.00	-	-	1.00
Vert. without model	-	0.83	-	0.96
Long. without model	-	-	0.92	0.98
Lat. with model	1.00	-	-	0.97
Vert. with model	-	0.76	-	0.90
Long. with model		-	0.94	0.98

Table 5.20 Summary of the normalised intensity errors between target and achieved motions

Signal	**Single-axis White Noise Signal**			**Three-axis Kalamata Signal**
Axis	**Lateral**	**Vertical**	**Longitudinal**	**Three-axis shake**
Lat. without model	0.97	0	0	1.19
Vert. without model	0.01	0.98	0.04	1.43
Long. without model	0	0	0.77	0.96
Lat. with model	0.96	0	0	1.01
Vert. with model	0.01	0.99	0.03	1.21
Long. with model	0	0	0.79	0.87

When considering single-axis white noise input to the platform, the values in tables 5.19 and 5.20 show that a reasonably good match was achieved for all the tests in all three directions. In particular, an almost perfect platform reproduction of target motions in the lateral direction was achieved. For other two directions the time history match was not so good, and in particular the values of the spectral ordinates of vertical signals and the intensity of longitudinal motions were poor. Generally, cross-coupling was low in all the tests; the largest value of 4% of the input intensity was observed for the bare platform in the vertical direction while exciting the platform in the longitudinal direction.

From the three-axis Kalamata earthquake tests it appeared that, although an excellent set of frequency components was maintained in all three translation axes, the intensities of the motions reproduced in the platform were generally higher than the respective targets. The largest intensity gain factor observed was 1.43 for the vertical motions. The presence of the test specimen had only a slight detrimental effect on the results, as can be seen in both tables.

Overall, the software control system performed well, at least in the frequency range of interest (0 - 20 Hz), when considering target and achieved platform displacements. This is shown by the good performance indices presented in tables 5.19 and 5.20 (values of around 1). However, resonances in the range of 16 to 17 Hz, in both vertical and longitudinal directions, could not be compensated by the software control system for several reasons which are discussed below.

The software control procedures at LNEC take into account the displacements and accelerations of the actuator pistons with a software algorithm using a non-square frequency response function matrix (§6.3.1). However as the transducers do not measure the translational motions of the shaking table platform directly this control procedure meant that some differences arose between different axes because there was a mechanical system between them comprising steel rods, cylindrical brackets (to fasten the actuator pistons to the steel rods), and bearings (connecting the rods to the platform).

The transfer functions between controlled accelerations (i.e. the actuator pistons) and the three translation axes of the platform were analysed in order to determine whether flexibility of the rod-bearing-bracket system could explain those resonances. It was confirmed that the differences between target and achieved platform motions could be explained in this way. A relatively low resonance frequency (16 to 17 Hz) associated with high damping was also observed in the transfer functions between actuators and platform. This can also be explained by the presence of the bearings and/or the cylindrical brackets. Again, in the vertical direction, a compression stress state on those elements, due to dead load, could explain a lower damping value than for the other directions.

The measured cross-coupling between translation axes, for adapted and non adapted tests, was very high. Typically, in terms of displacements components, for an excitation in a given direction, the output translation intensities observed in the other directions were less than 4% of the input. However, when considering the acceleration components for the same motions, the intensity of cross-coupling between axes was much greater. In fact, the response spectra and FRFs of achieved motions clearly show that the resonances of the rod-bearing-bracket systems, already mentioned, were the main source of cross-coupling between axes.

5.5.5 Further tests at the Bristol site

The initial programme of tests performed on the four shaking tables, described above, was supplemented by a further series of tests performed on the Bristol table in July 1997. These tests investigated, in particular, some of the key issues that were highlighted by the tests previously performed at the four laboratories, and compared the performance of the old analogue controller at Bristol with the new DARTEC 9600 digital controller that was installed towards the end of this research. Additional tests were performed to investigate some new control techniques currently being developed at Bristol which should cope with non-linear specimen performance. Tests were also performed to confirm the adverse effect of rotational components on specimen behaviour.

5.5.5.1 Performance of the DARTEC 9600 digital hardware control

Characterisation tests of the shaking table performance with the new digital hardware controller show that the oil column resonances at Bristol remain an issue. With the old analogue controller the natural frequencies in the horizontal and vertical axes were 16.25 Hz and 23.0 Hz respectively (table 5.8). Tests with the new controller (§4.3.2.6) showed slightly different frequencies of 15.0 Hz and 23.13 Hz in the same axes. The amplitude of the resonances were also very similar. These results, as expected, are very similar because both the old and the new controllers are relying mainly on displacement feedback to close the control loops. The new digital controller does, however, have many advantages over the old analogue controller, one of which being that it can be modified in the future to incorporate more advanced real-time control techniques (§6.7). Several different options (§6.2.1) are currently being considered with the aim of reducing the oil column resonance in the near future.

5.5.5.2 Non-linear iterative matching methods

After the results obtained with the non-linear iterative matching techniques at ISMES and LNEC, additional tests were performed at Bristol using non-linear iterative matching techniques. These were found to be more stable than the linear iterative matching normally performed at Bristol, confirming the advantages of this type of matching. However, the use of a new real-time control technique, the Minimal Control Synthesis (MCS) algorithm (Stoten, 1993) (§6.7), is beginning to supersede such iterative matching methods. MCS has proved to be very effective at controlling the Bristol shaking table, although currently only for platform motions with a low frequency content. Several of the most recent tests at Bristol have therefore been performed using real-time MCS control rather than iteratively matching the time histories in the normal way with the software control system. In one recent test MCS was able to control the platform motion with a very large, lightly damped specimen on the platform. Without MCS, such a specimen would have interacted significantly with the table and this would have resulted in undesirable platform rotations, making such a test very difficult to perform.

5.5.5.3 Frequency response tests with MCS

While MCS has proved to be very good at controlling low frequency input motions at Bristol and ISMES (§6.7), this system for controlling shaking tables still needs further development to allow it to be used across the full operating frequency range of the table. Figure 5.55a shows the transfer function of the X axis of the Bristol table with the 5 tonne flexible specimen mounted on the platform. In this case only the digital hardware controller is being used to control the platform motion. Figure 5.55b shows the change in frequency response of the table when MCS is also used to control the platform motion. The amplitude of the oil column resonance is somewhat reduced when the MCS controller is running, but there is otherwise little difference in the performance of the table. The table performance at high frequencies does not significantly improve with MCS because of relatively low resolution of the displacement transducers and the difficulties of accurately converting the analogue signals into the digital values used by MCS to monitor the behaviour of the table. These problems are described in detail in §6.6. As part of the development of MCS, work is currently taking place to develop composite filters which can combine displacement and acceleration signals to form an equivalent high resolution displacement signal. In the future these filters may significantly improve our ability to measure and control platform motions accurately over a much wider range of frequencies.

5.5.5.4 Bearing compliance

The potential effects of bearing backlash and their flexibility on the performance of a shaking table were discussed in §4.3.2.4. After the tests at the four laboratories, additional tests were performed at Bristol to investigate the extent to which these problems might be limiting the maximum possible performance of any shaking table control system. This limitation will occur because any compliance of the bearings is not measured as part of the feedback control loops of the table and is therefore uncontrolled.

Accelerometers were attached across the bearings at each end of one of the vertical actuators at Bristol. The transfer functions at each of these points were measured with respect to a random noise drive signal applied to the table. Ideally, if the bearings are infinitely stiff, the transfer functions on either side of the bearing connected to the table platform are identical, and the transfer functions on either side of the bearing connected to the reaction mass should be zero. Figure 5.56 shows the transfer functions measured at these locations and table 5.21 summarises these results.

From table 5.21 it can be seen that there is a significant axial flexibility in the bearings in the Bristol table. Of the total platform response, only about 70% is developed across the actuator itself. Fortunately this deficiency is caused by axial tension and compression of the bearings rather than bearing backlash (the transfer functions shown in figure 5.56 are not very noisy, even at high frequencies). Therefore, although the displacements recorded by the actuators may not be completely representative of the actual platform motion, if external instrumentation is used to monitor the achieved platform motion in any iterative matching process (§6.3.1), these linear errors can be effectively controlled.

Table 5.21 Summary of the FRFs across actuator bearings

Position	Frequency (Hz)	Amplification
Platform (top of upper bearing)	23.125	31.16
Top of actuator (bottom of upper bearing)	22.750	26.16
Bottom of actuator (top of lower bearing)	22.375	4.17
Reaction block (bottom of lower bearing)	23.250	0.99

5.5.5.5 Kinematic model

The kinematic model of a shaking table is the definition of how movement of the actuators controls the actual movement of the platform, assuming that there are no other dynamic effects. In a single-axis table the kinematic model is simple; the displacement of the platform being the same as the displacement of the actuator. The kinematic model for a shaking table that can move in more that one axis is more complicated. An example of the coupling between two actuators in a two-axis table is shown in figure 5.57. This figure shows the extension of the horizontal actuator that must occur if the platform is to move vertically upwards without any additional horizontal motion. The equation shown for the additional extension of the horizontal actuator is an approximation based on the assumption that the length of the actuator is much larger than the possible vertical movement of the platform. If this approximate formula was used to calculate the kinematics of the Bristol table, the resulting error for maximum vertical movement is about 0.09%. In the LNEC table, which has much longer actuator/extension rod systems, the approximate formula only results in an error of about 0.04%. In most older tables, the kinematic model is built into the analogue hardware controller and the approximate formulation can calculated using analogue multipliers and dividers. In the newer digital hardware controllers the exact formulation of the kinematic model can be carried out (Stoten and Gomez, 1998), eliminating even these small errors. However, in order to perform either calculation, accurate measurements of the as-built locations of the ends of the actuators are needed, ideally at sub-millimetre accuracy. In Bristol, the co-ordinates of the centres of the actuator bearings were re-measured after the installation of the new digital controller, using a MONMOS total station, to an accuracy of ± 0.5 mm. Based on the information from this survey a much more accurate kinematic model has now been implemented, virtually eliminating any cross-coupling of table axes under static loading.

5.5.5.6 Additional tests with flexible model

A few additional tests were also performed with the flexible model on the platform confirming the adverse effect of rotational components on specimen behaviour. The ability of MCS to compensate for these rotational motions was also investigated. Figures 5.58 and 5.59 show the behaviour of the platform and the flexible specimen with and without MCS running. The second plot in figure 5.59 shows a significant improvement in the longitudinal platform response when MCS is running compared with the equivalent plot in figure 5.58. The specimen response is shown in the third plot in each figure. The maximum specimen response is clearly much larger when the desired motion is more accurately reproduced. However, the rotational accelerations (recorded as the difference between vertical accelerations at the two edges of the platform) are also higher when MCS was running. These results show that MCS is effective in controlling platform motions at Bristol with a low frequency content (i.e. displacements), but that control of the higher frequencies caused by specimen-table interaction, and characterised by very small displacements, requires the use of composite filters which combine acceleration and displacement signals (§6.6). If the rotational motions were controlled then the specimen could be expected to show an even higher response.

In an attempt to find methods of increasing the stiffness of the vertical actuators and reducing the oil column resonances without resorting to mechanical means (§6.2.1), the stiffness of the vertical actuators and the ease with which the Kalamata earthquake could be matched were investigated with different levels of preload oil pressure. The centre position of the platform was also adjusted to see whether running the platform with the vertical actuators almost closed or almost fully extended could reduce table-specimen interaction.

The preload pressure, at Bristol, is normally only used to offset the static mass of the platform and specimen, and tests showed that use of too high or low a preload pressure had an adverse effect on the stiffness of the vertical actuators and increased table-specimen interaction. When the preload section of the vertical actuators was not carrying the static mass of the platform and specimen properly, the dynamic capacity of the vertical actuators was being used to compensate for the error in preload pressure. It is suspected that this forced the hardware controller to work slightly harder when controlling the extensions of the actuators, resulting in the loss of performance. Additional work could be done to confirm the actual reason for the adverse effect of a badly balanced table under static loading (i.e. wrong preload) on the dynamic performance, but this is probably unnecessary – it should simply be noted that any system that is used to counteract the static mass of a platform and specimen should be adjusted to do so as accurately as possible. This avoids a degradation in table performance, and will also allow the vertical actuators to excite the platform to their dynamic limits, rather than wasting their dynamic capacity carrying the static load. Tests also showed that similar problems with a loss of performance occur if the forces in the horizontal actuators are not all zero under static load (i.e. when the actuators are fighting, §4.2.4). Accurate calibration of the displacement transducers in the actuators helps to avoid this problem, and fine adjustment of the centre positions of each actuator can be used to stop any fighting. An alternative method of avoiding actuator fighting, called force balance compensation by MTS, uses feedback from load cells or differential pressure cells in the actuators to adjust the forces in all the actuators to zero under static loading.

The tests that investigated the effects of running the platform with the vertical actuators almost closed or almost fully extended produced some interesting results. It was found that running the platform in either a high or low position reduced table-specimen interaction by about 15%. It is currently unclear as to why the static position of the platform should affect the table performance and reduce table-specimen interaction. There is a need for further study of this effect. This will also allow a decision to be made as to whether this effect can be used in practice to improve table performance during testing.

The performance assessments at all the sites were all limited to rigid and elastic payloads – studies of the effects of non-linear payload responses, representative of failing specimens, are the next essential requirement for all the tables because these will provide a worst case scenario for table control hardware and software. Continuation of the research into the performance of shaking tables with non-linear models is currently in progress, and initial tests of the MCS system with non-linear table performance have been performed (§6.7), with very positive results.

5.5.6 Recommended performance assessment procedures

One of the primary aims of this research was to develop a systematic methodology for regularly assessing the performance of an earthquake shaking table. The methodology adopted as part of this research has proved, on the whole, to be workable and effective. It has produced a valuable benchmark database against which the performance of the four shaking tables can be assessed in the future. The test programme adopted at the start of these investigations was extensive, and could reasonably be reduced in the light of experience for any future testing. In addition, any regularly-used assessment procedure should take into account the particular features of the shaking table in question. Accordingly, the procedure outlined below sets out the broad objectives of the assessment process.

The assessment of a shaking table should be carried out in two stages. The initial stage should be systematic and rigorous, with the aim of developing a comprehensive benchmark database for the shaking table in question. In developing this database, the operators will learn valuable lessons with respect to the performance characteristics and limits of their shaking table. The second stage of the assessment procedure should be carried out periodically to monitor the performance of the shaking table. For example, it should be used after major maintenance or as part of an annual calibration. The second stage should be a sub-set of tests from the first stage.

The recommended outline assessment procedure is as follows.

<u>Frequency response functions</u>

- Evaluate the frequency response function of the shaking table for each axis, including cross-coupled axes (e.g. longitudinal with pitch) for a bare platform, 50% rigid payload capacity and 100% rigid payload capacity. Use either broadband random noise or repeated impulse input signals. Repeat with different input signal gain levels to calibrate the system gains up to maximum performance.

- Evaluate the frequency response function of the shaking table for each axis, including cross-coupled axes (e.g. longitudinal with pitch), for flexible specimens of 50% payload capacity and 100% rigid payload capacity. The specimen natural frequency and centre of gravity should be chosen to suit the performance characteristics of the shaking table. Use either broadband random noise or repeated impulse input signals. Repeat with different input signal gain levels to calibrate the system gains up to maximum performance.

Time history response

- Generate three independent broadband artificial time histories having frequency components compatible with the operating bandwidth of the shaking table. For example, a frequency-scaled response spectrum based on the Eurocode 8 design spectrum might be used as a basis of the artificial time histories.
- Evaluate the similarity of the time history response of each axis by applying one component of the artificial time history to each axis in turn. The tests may be done with a bare platform. Compare the achieved acceleration and displacement time histories and the achieved acceleration response spectra with the required motions. Optionally, repeat these tests at different gain levels to calibrate gains.
- Evaluate the time history response of the shaking table with multi-axis (usually translational degrees of freedom only) artificial time histories for bare platform, 50% rigid payload and 100% rigid payload. Compare the achieved acceleration and displacement time histories and the achieved acceleration response spectra with the required motions. Optionally, repeat these tests at different gain levels to calibrate gains and assess effects of rigid payloads.
- Evaluate the time history response of the shaking table with multi-axis (usually translational degrees of freedom only) artificial time histories for flexible specimens of 50% payload capacity and 100% rigid payload capacity. The specimen natural frequency and centre of gravity should be chosen to suit the performance characteristics of the shaking table. Compare the achieved acceleration and displacement time histories and the achieved acceleration response spectra with the required motions.

Measure cross-coupling effects between axes, especially coupling of translational degrees of freedom with associated rotational degrees of freedom (e.g. longitudinal with pitch). Optionally, repeat these tests at different gain levels to calibrate gains and assess effects of rigid payloads.

Appropriate sub-sets of the above tests can be selected for periodic performance assessments and routine pre-test calibration. The selection will depend on the characteristics of the particular shaking table.

5.6 Key issues identified in the study

5.6.1 General

For the first time, the performances of four different earthquake shaking tables have been rigorously compared using common procedures. Each of the four shaking tables has been shown to produce high fidelity platform motions for a wide range of test conditions. Acceleration and displacement time histories were reproduced with fairly small errors, subject to the performance limitations of the tables. This important finding strengthens the reliability of experiments carried out on the four tables and consolidates the international status of the four laboratories. European researchers now have access to four shaking tables of proven high quality.

Detailed characterisations of the dynamic performances of the four shaking tables have been produced. In the course of this, relevant software has been validated and the strengths, weaknesses and necessary enhancements of the four tables have been identified.

A viable, general, systematic methodology for assessing the performance of shaking tables has been developed and proved. This methodology can be used as a basis for developing specific procedures for any particular shaking table.

This research has resulted in a very successful interaction between the research staff at the four laboratories. Regular exchange visits enabled the staff to observe the procedures at the collaborating laboratories and gain a wider perspective of shaking table experimentation. A free exchange of ideas, software, documents and procedures has also established a strong working relationship between the four laboratories, and also with other European Union researchers who had access to the shaking tables through the European Consortium of Earthquake Shaking Tables (ECOEST).

The following sections summarise the key specific conclusions of the testing programmes at the four laboratories resulting from this PhD research.

5.6.2 Hardware control systems

The frequency response tests at all the laboratories showed the importance of following a methodical and efficient hardware tuning routine. Slight variations in the individual actuator servo-controller gains lead to significant and sometimes severe resonances and instabilities. For each table, other than at Bristol, careful adjustment of the feedback parameters in the hardware controller produced acceptably flat frequency responses for rigid payloads. The frequency response curves for the flexible payloads, on the other hand, showed peak and notch effects. The more sophisticated Athens and ISMES and LNEC hardware, incorporating many feedback loops, was better able to compensate for these effects, but none of the tables could be adjusted to achieve a flat frequency response curve for all the load cases tested. In Bristol, the difficulties with oil column resonances mean that a rigorous hardware tuning is not performed before each test, as happens in the other laboratories, and the software control system is used to compensate for the errors in the table tuning.

The extensive and systematic performance tests that formed part of this research have produced an important benchmark database against which the future performance of the four shaking tables can be compared. For example, after major maintenance, a sub-set of the performance tests can now be run and the results compared with the benchmark to ascertain whether the system performance has changed. Appropriate remedial measures can now be undertaken with more confidence.

This research has demonstrated the value of carefully tuning the hardware control system of a shaking table before each test to optimise the system transfer function. The broad range of loading conditions explored as part of this research has also given the operators at each laboratory greater understanding of the kind of adjustments that should be made for different types of tests. Extensive application of various tuning techniques developed during these tests has also enabled a better assessment of the effects and limitations of a single tuning during a series of tests.

Comparison of the tuning procedures adopted by each of the laboratories gave the opportunity of assessing the value of different methodologies and instrumentation for obtaining objective and traceable evaluation of the tuning results. At the start of these tests, the use of single-axis random noise to generate the frequency transfer functions of all the axes was only practised at Bristol. The other three laboratories had previously used sine-sweep tests to tune the control hardware. The use of random noise signals, which produce results significantly faster than sine sweep tests, is now practised at all four laboratories.

The extensive application of tuning operations with different payloads allowed the first quantitative evaluation of the acceptable amplification factors in the frequency transfer functions of any axis of a shaking table. Factors larger than 2-3 may result in poor coherences and difficulties with the stability of the software control system after a few iterations.

Good maintenance of the table, both of the mechanical parts and of the electronic ones, is of prime importance in order to achieve a good system tuning. For example, any backlash in the bearings at the ends of the actuators will significantly reduce the overall performance of the system, and it will not be possible to tune the hardware control system as effectively.

5.6.3 Software control systems

After considering many different methods of tuning the four shaking tables as well as the ability of the software control systems to compensate for significant errors in the system transfer functions, it became clear from the results of the tests performed during this project that the best reproduction of time histories responses on the shaking tables was obtained when the system transfer function was as flat as possible, with the response of the platform equal to the input signal at all frequencies.

It was found that each of the four tables could reproduce the required motions with the platform bare or with rigid payloads. For each input record, either a good acceleration match or a good displacement match could be achieved, but usually not both at the same time. With the Athens and Bristol shaking tables it proved possible only to match accurately either the acceleration time histories or the displacement time histories for a given motion. Simultaneous control of the platform displacement in the case of an acceleration match, or platform acceleration in the case of a displacement match, proved difficult to achieve. Future extension of the control algorithms at these two sites to combine displacement, velocity and acceleration performance so as to produce a composite control signal that encompasses their full operating frequency range is currently planned. Such a system yields better simultaneous matching of accelerations, velocities and displacements, as shown by the results obtained at ISMES and LNEC. To achieve a good response spectrum match it was essential to achieve a good acceleration time history match. Matching displacements did not accommodate the higher frequencies present in the acceleration signal, which in turn controlled the detail of the response spectra. In the case of the flexible specimen, the tuning and the drive signal compensation had to be more rigorous if the effects of table-specimen interaction were to be minimised to acceptable levels.

When the specimen with a well defined natural frequency of about 6 Hz was placed on any of the shaking tables, the current control systems could not reproduce the required platform motion as accurately as when there was no specimen on the platform. Although it was not possible to test the ability of the control systems to cope with a specimen that became non-linear during a shake, it can be assumed that the results would be worse than those recorded with the linear specimen on the platform The current control systems are completely passive and no adjustment of the table tuning or the drive signal took place during a seismic test. If the specimen characteristics change during a seismic test, the specimen's interaction with the table will significantly effect the platform motion. Unfortunately this type of non-linear specimen response is what occurs in many shaking table tests, and unless it is possible to prepare and test many specimens as part of the iterative matching process the error between the required and achieved platform motion may cause difficulties.

The effects and limitations of each control parameter (e.g. use of an appropriate frequency range of the drive signal for the best displacement control) and methodologies for the correction of the errors (in the time or frequency domain) were determined. The importance, for accurate reproduction of platform motions, of the adopted control modes (e.g. acceleration or displacement control) and of the number of the axes controlled was also studied.

The importance of the effects of the undesired motions was shown by the response of the flexible specimen in the various tests. This problem can affect a test result not only from a purely scientific viewpoint but can also directly affect the technical acceptability of the test results. For example, this is especially important in the case of the qualification of equipment intended for the safety of nuclear power plants or for electrical transmission (De Silva, 1983). At present the relevant standards for the qualification of electrical equipment by the time-history method (such as the widely used IEEE Std. 344-1987) provide only general requirements for rotational motions, without any reference to acceptable values. Reasonably severe limitations on lateral undesired motions are given, and acceptance criteria for the enveloping of the Required Response Spectrum are specified. With reference to these requirements, the results obtained on all four shaking tables show errors that are well within the acceptable requirements. It is obvious, however, that the consequences in terms of structural reliability of the equipment being tested and of the functioning of the electrical devices may be totally different if the specimen accelerations are 4 or 5 times higher than those recorded in a test when rotational motions are uncontrolled. Finally, it must be stressed that the results of the tests detailed in §5.5.3.2 show that unwanted motions caused by table-specimen interaction do not in practice act as an additional excitation source which would give rise to a conservative test. In fact the rotations seem to act as energy absorption mechanisms, which, in practice, reduce the response of the specimen. Therefore, not only must the global amplitudes of the undesired motions (the rotations in particular) be assessed but, most importantly, their frequency content must be carefully determined, as this has great influence on the interaction between the specimen and the table.

One problem identified during this research was the need to record accurately the very small displacements and accelerations that often occur on a shaking table at the start and end of the reproduction of an earthquake record. This is particularly important in the case of acceleration matching, where the accelerations at the end of the signals are very small. These accelerations must be measured very accurately in order to improve the ability of the software algorithms to compensate for any errors between the specified and acquired signals at these small amplitudes. In Bristol, this required a modification of the acquisition software to incorporate the user programmable gains that were used at the other three sites, which resulted in much-improved low-amplitude platform responses.

5.6.4 Major enhancements to the shaking tables

5.6.4.1 Bristol

At the time the initial tests were performed, the Bristol shaking table was difficult to tune, partly because the hardware did not allow easy modification of the feedback loops and partly because several forms of feedback control were not implemented (e.g. force, pressure and velocity feedback). For these reasons the Bristol shaking table took a long time to calibrate. Over the period 1994-1997, the software control procedures were gradually changed where possible, and the control hardware was re-calibrated as much as practicable. Following these tests, which highlighted the difficulties of achieving flat response spectra at Bristol, a new digital hardware controller was installed in the shaking table system, early in 1997. While this new controller has not in itself significantly improved the dynamic performance of the table, mainly because no additional feedback loops have been implemented, it is now allowing the development of some new active-control software (§6.7) which effectively adjusts all the existing feedback gains in the control system in real-time; and this is beginning to produce noticeable improvements in table control in the non-linear performance range of the table (§6.7).

5.6.4.2 Athens

The analogue hardware control system in Athens, although offering good control, is now more than ten years old and the electronics are deteriorating to such an extent that they require constant maintenance. The Athens laboratory is therefore in the process of upgrading this hardware with a new digital MTS system. This new state-of-the-art system will provide all the controls that were present in the older analogue system (*MTS 1985*) while incorporating some additional adaptive systems that can only be implemented in a digital control system. A real-time control system (§6.7) is also currently being implemented as an upgrade to the existing control software.

5.6.4.3 ISMES

The shaking table in ISMES is controlled by an old MTS analogue hardware system but the software control has recently been upgraded. This new software produces some excellent results when used properly, as can be seen from the tests at ISMES (ISMES, 1996). However, a real-time control system (§6.7) is currently being implemented as an upgrade to the existing control software to allow accurate control of the table during the testing of non-linear specimens. There are no plans to upgrade the hardware at ISMES in the near future.

5.6.4.4 LNEC

The shaking table in LNEC, being the most recently constructed, is already controlled by a digital hardware system with sophisticated control software. In addition to this software, a real-time control system (§6.7) is currently being implemented as at the other three sites.

5.7 Conclusions

From the results of the tests at the four laboratories the following main conclusions can be drawn:

- Perfect tuning of the hardware control system of a shaking table is impossible. However, good results can currently be achieved although such tuning is often difficult and depends a great deal on operator experience.
- Good tuning is necessary, but is not on its own sufficient to guarantee good results. An efficient software control system is essential, especially for the compensation of non-linearities and cross-coupling between axes.
- Better tuning of the hardware control system simplifies the extent to which the software has to compensate for inaccuracies in the platform motion and so makes desired test motions easier to achieve.

- Best overall results were achieved when the platform displacements were actively controlled, as this also resulted in a reasonably good match of the required accelerations. However, in this case, it is likely that some of the high frequency detail in the acceleration signals will be lost.
- If very accurate control of platform accelerations, or platform response spectra, are required then the platform accelerations should be actively controlled, but this will be at the expense of good reproduction of the desired platform displacements.
- Matched drive signals are generally not transferable between different table axes, or scaleable to different amplitudes, without a detrimental effect on the accuracy of platform response.
- All axes should be actively controlled by the software control system even if this only means controlling the motions in these axes to zero.

The sets of tests at the four facilities described in this chapter also highlighted the need for a detailed investigation into the effect of inaccuracies in the achieved platform motion on the results of shaking table tests. The tests identified weaknesses in the control methodology of shaking tables, and the need for improved testing techniques to cope with testing of specimens that have significant dynamic interaction with the shaking table. Finally, the tests highlighted a need for new control techniques to deal with real-time control of shaking tables and non-linear specimen response during a test.

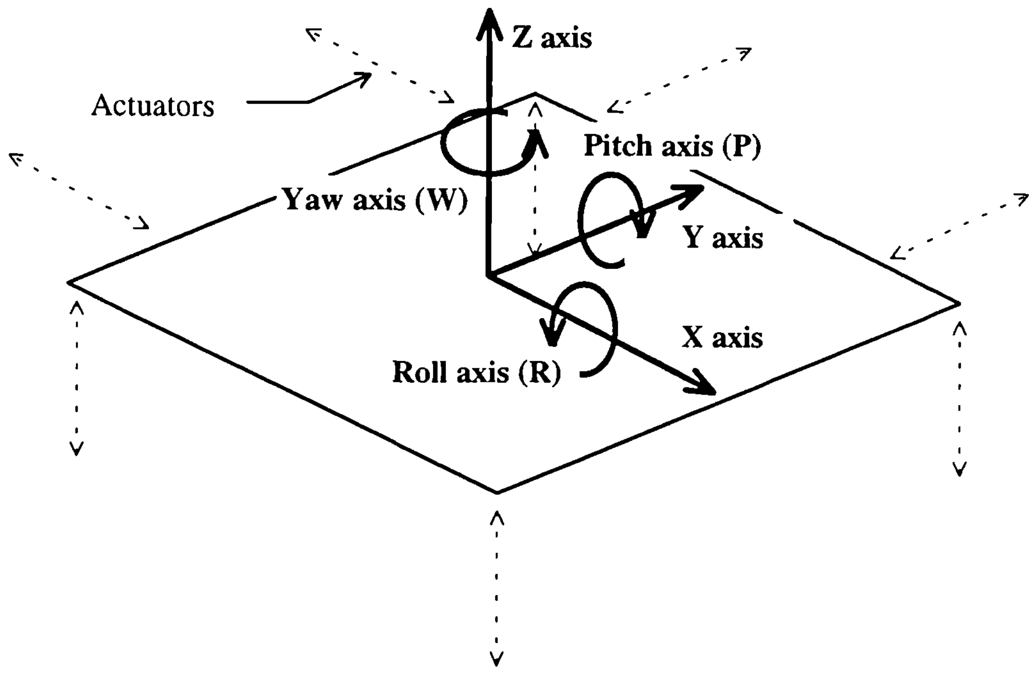


Fig. 5.1 Naming convention for shaking tables axes adopted in this thesis

(Original in colour)

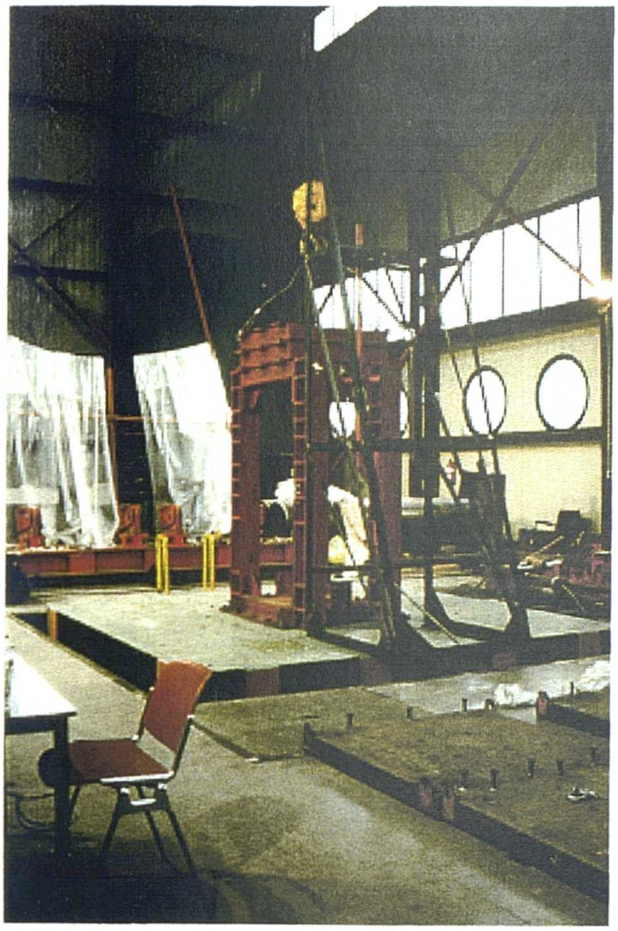


Fig. 5.2 The shaking table at the NTU Athens, Greece

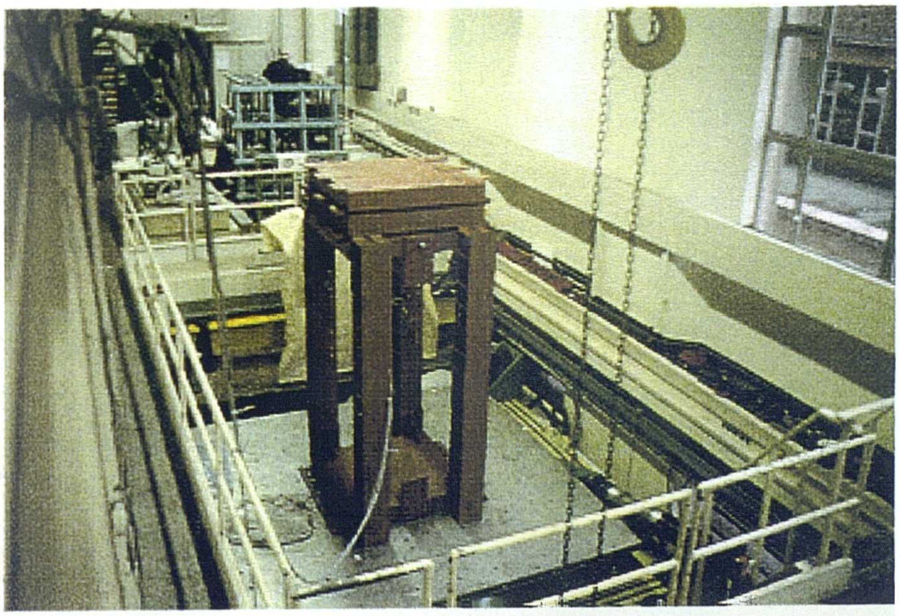


Fig. 5.3 The shaking table at Bristol University, UK

(Original in colour)

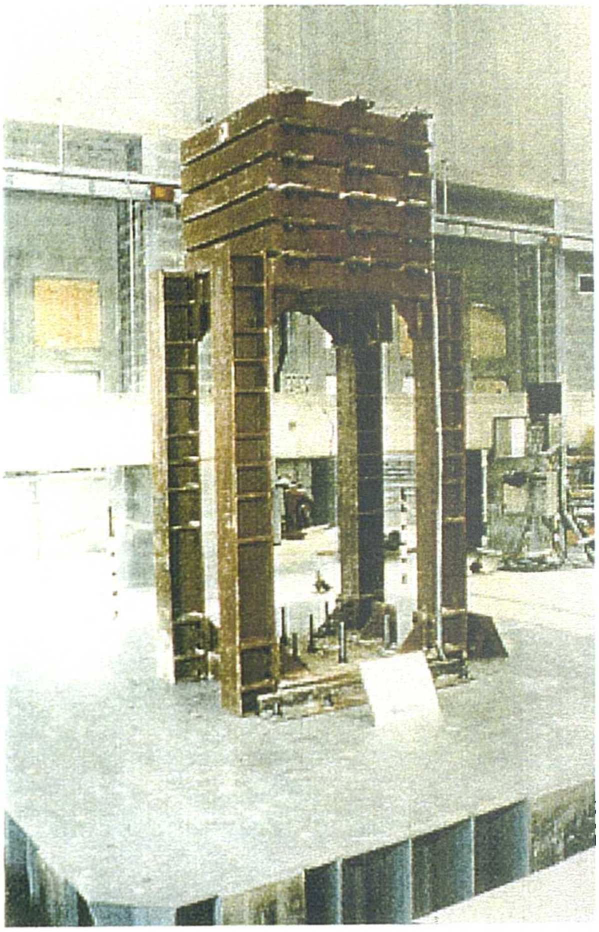


Fig. 5.4 The shaking table at ISMES, Seriate, Italy

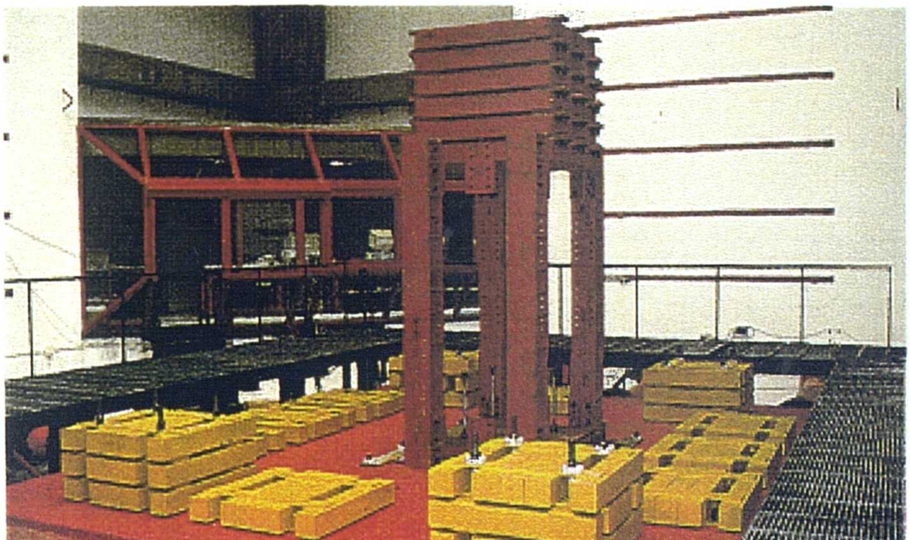


Fig. 5.5 The shaking table at LNEC, Lisbon, Portugal

(Original in colour)

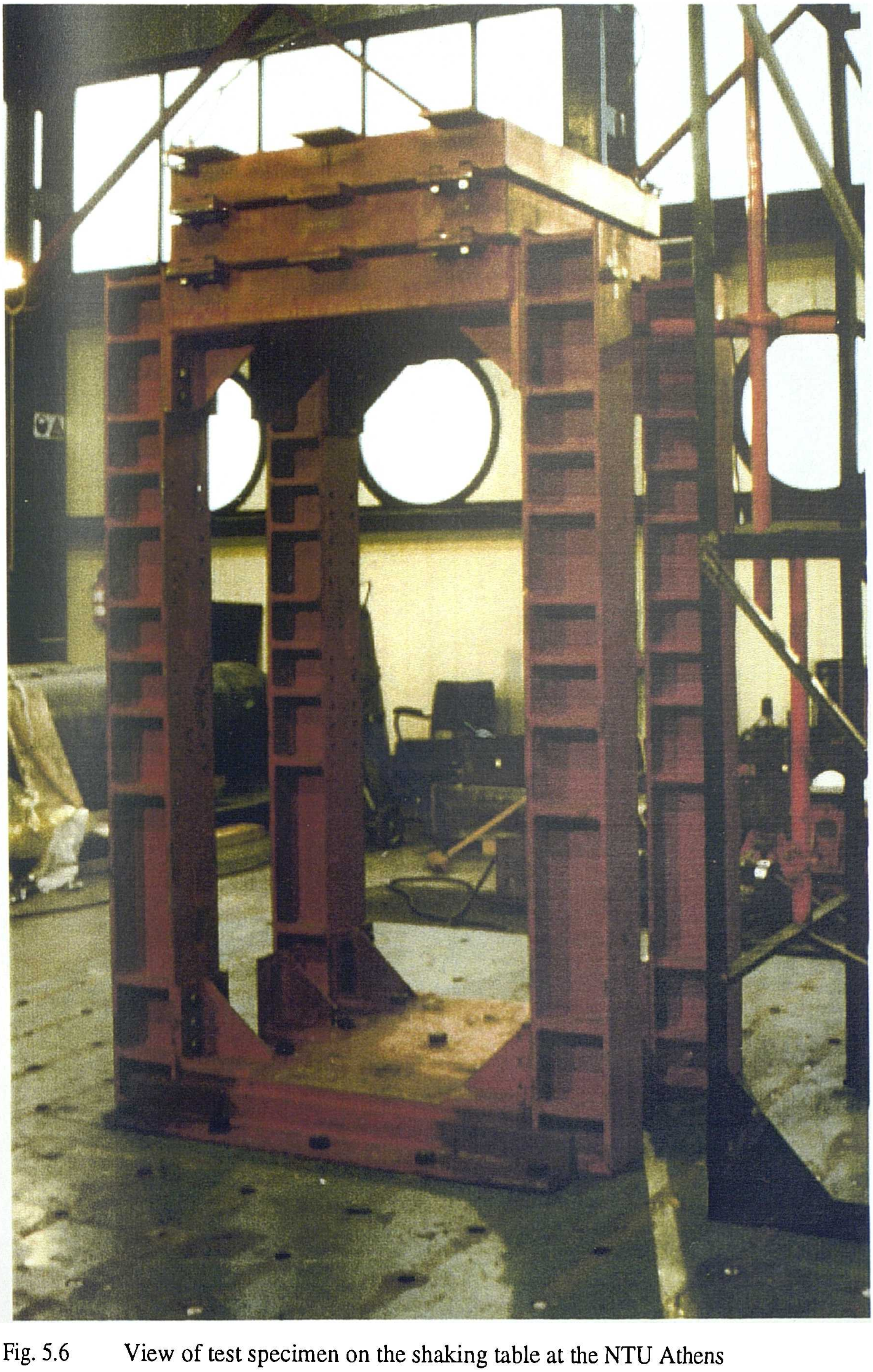


Fig. 5.6 View of test specimen on the shaking table at the NTU Athens

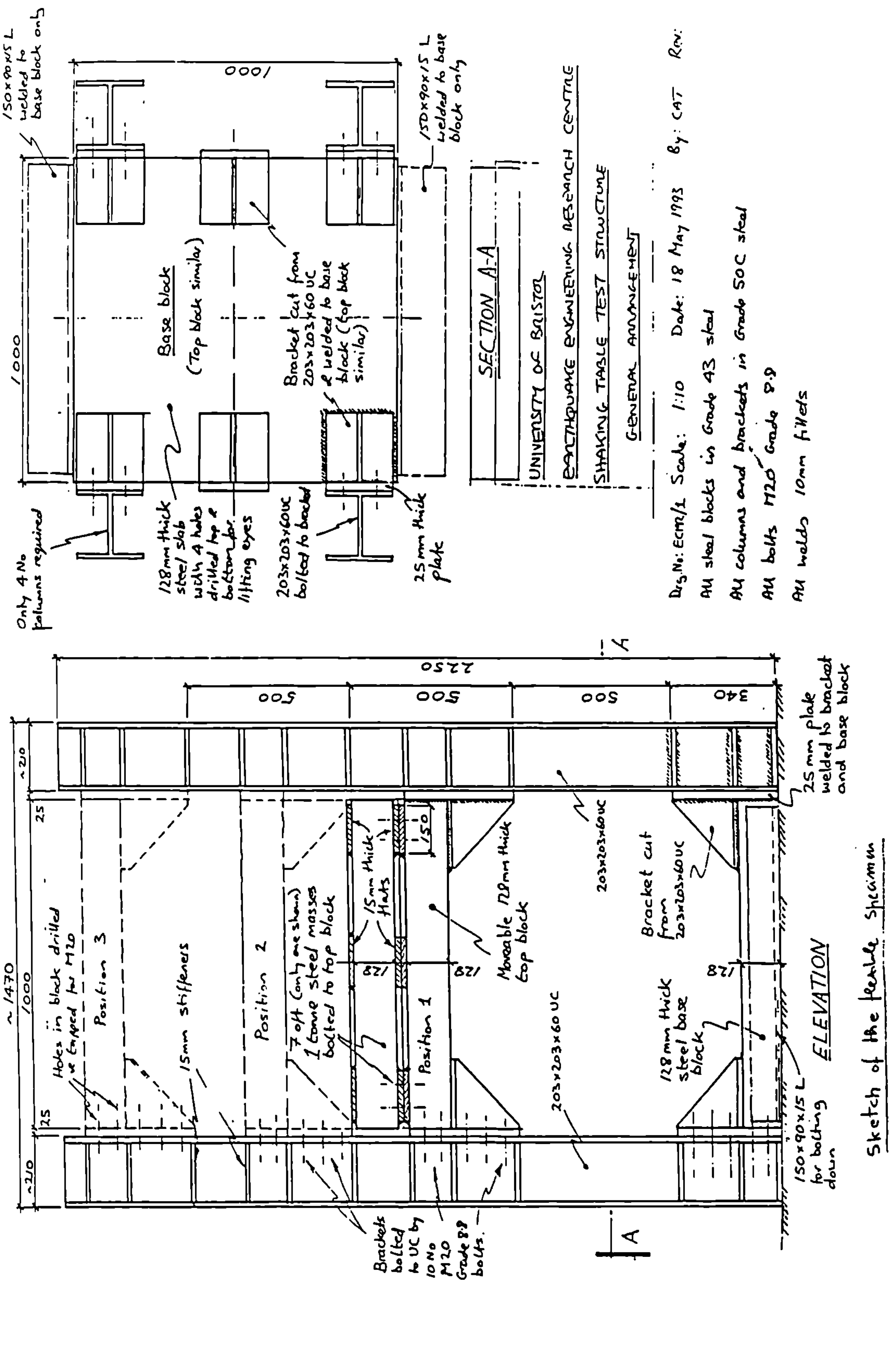


Fig. 5.7 Outline drawings of test specimen

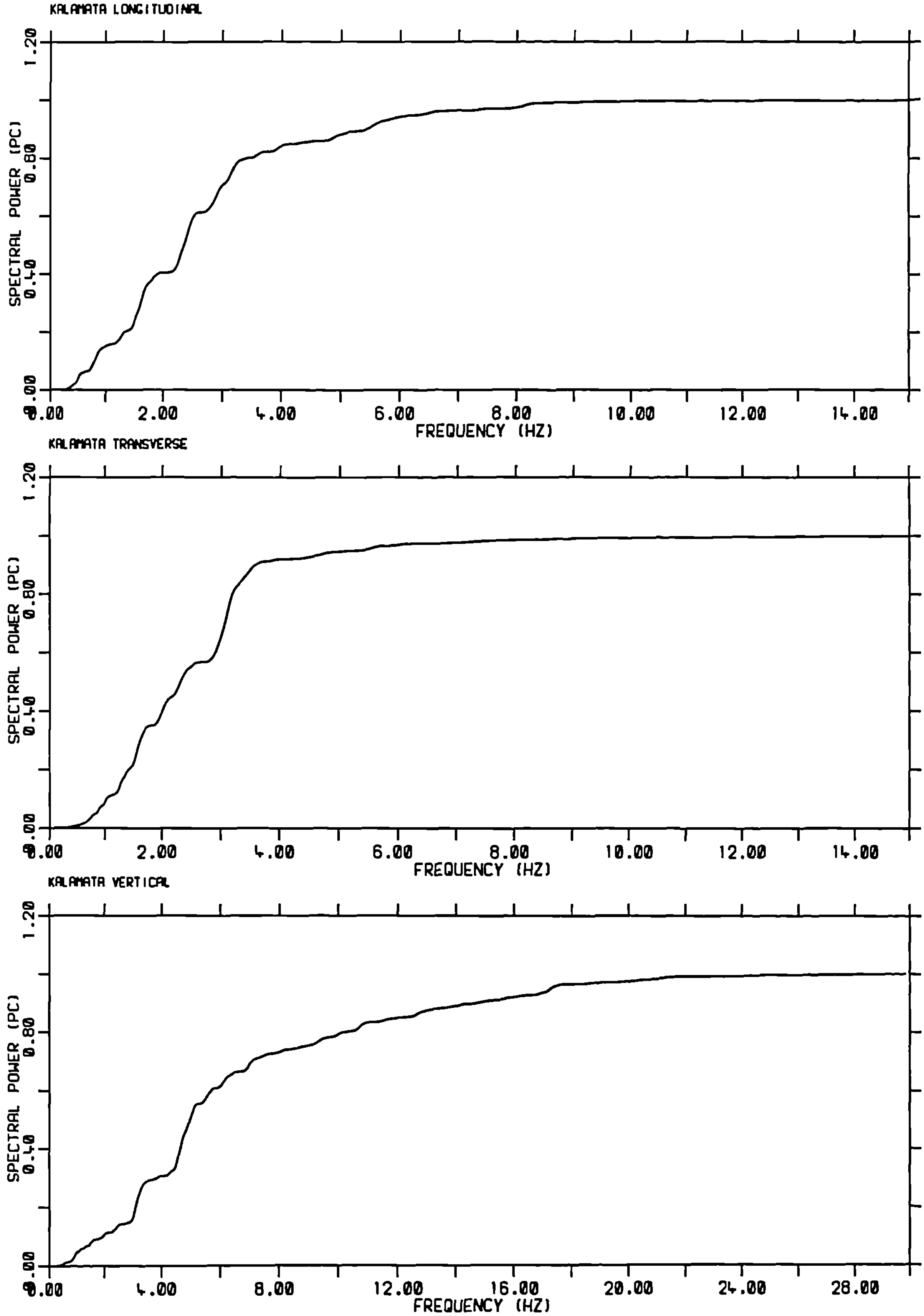


Fig. 5.8 Cumulative power spectra of the three axes of the Kalamata acceleration time history

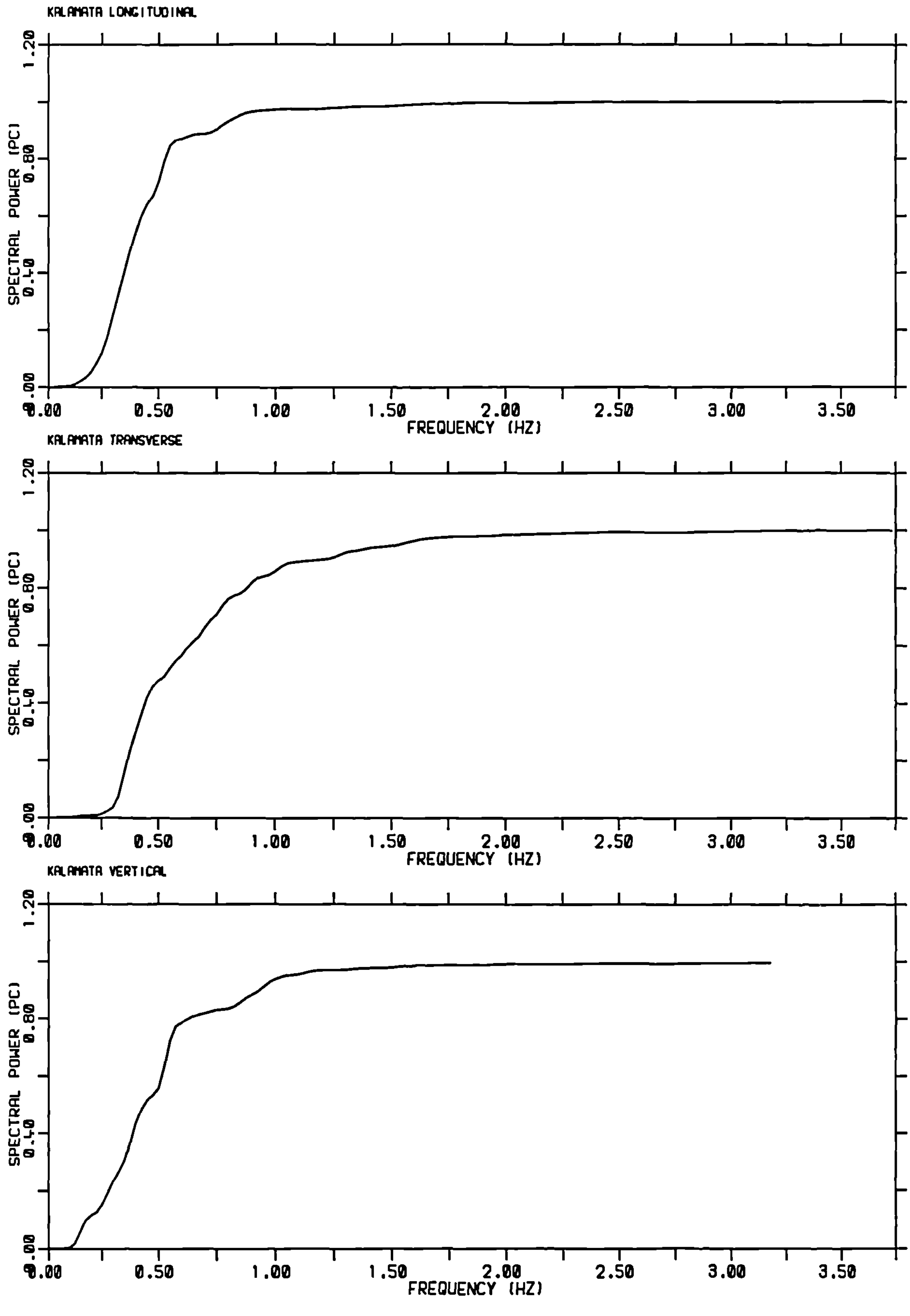
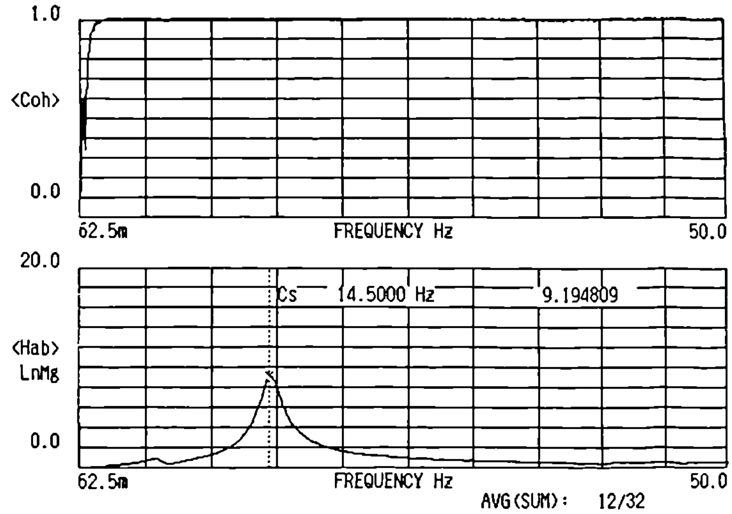
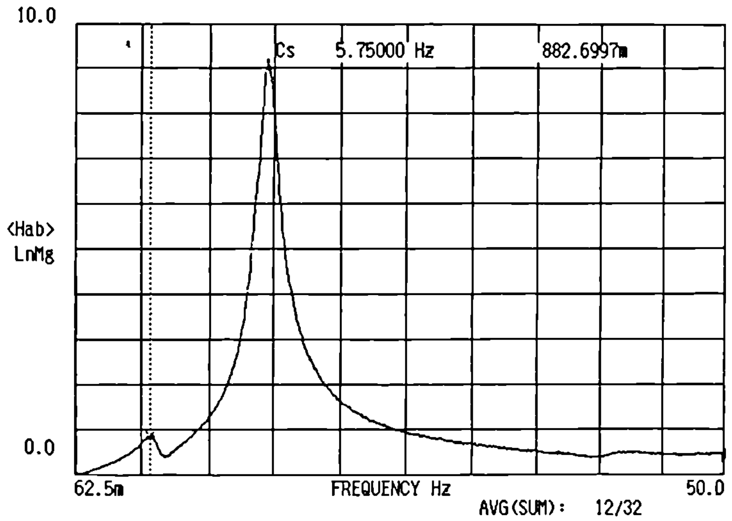


Fig. 5.9 Cumulative power spectra of the three axes of the Kalamata displacement time history

(details of the abbreviations on these figures and the calculation of natural frequencies and damping ratios from pole and zero data can be found in Appendix B.2).

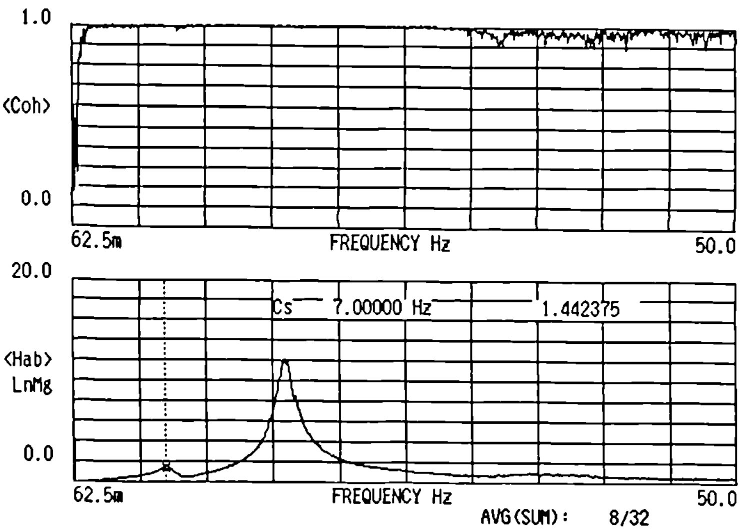




POLES 13				ZEROS 8		
1	66.0390m			21	2.87578		
2	1.67061	±j	5.78578	22	-3.50753		
3	-792.357m	±j	14.5477	23	1.49751	±j	5.29317
4	-13.5509	±j	20.0380	24	-9.30363	±j	36.1957
5	-6.69032	±j	34.7965	25	-2.83020	±j	41.3364
6	-2.12919	±j	42.0441	26			
7	-2.74234	±j	52.7066	27			
8				28			
9				29			
10				30			
11				31			
12				32			
13				33			
14				34			
15				35			
16				36			
17				37			
18				38			
19				39			
20				40			

Fig. 5.10 Frequency response function of the Transverse axis (Y) of the Bristol table with a 5 tonne flexible specimen mounted on the table

(details of the abbreviations on these figures and the calculation of natural frequencies and damping ratios from pole and zero data can be found in Appendix B.2).



	POLES 14				ZEROS 14		
1	-4.95769m			21	826.281m	±j	1.90044
2	-16.0921			22	-292.948m	±j	8.87643
3	-633.268m	±j	7.80079	23	7.85572	±j	12.2272
4	9.99338	±j	13.3650	24	12.7941	±j	21.5151
5	-737.327m	±j	15.9805	25	-16.8213	±j	29.2477
6	17.9842	±j	20.6201	26	-3.00561	±j	31.8349
7	-9.77018	±j	32.0628	27	23.5694	±j	42.0866
8	-3.66493	±j	33.8395	28			
9				29			
10				30			
11				31			
12				32			
13				33			
14				34			
15				35			
16				36			
17				37			
18				38			
19				39			
20				40			

Fig. 5.11 Frequency response function of the Longitudinal axis (X) of the Bristol table with a 5 tonne flexible specimen mounted on the table

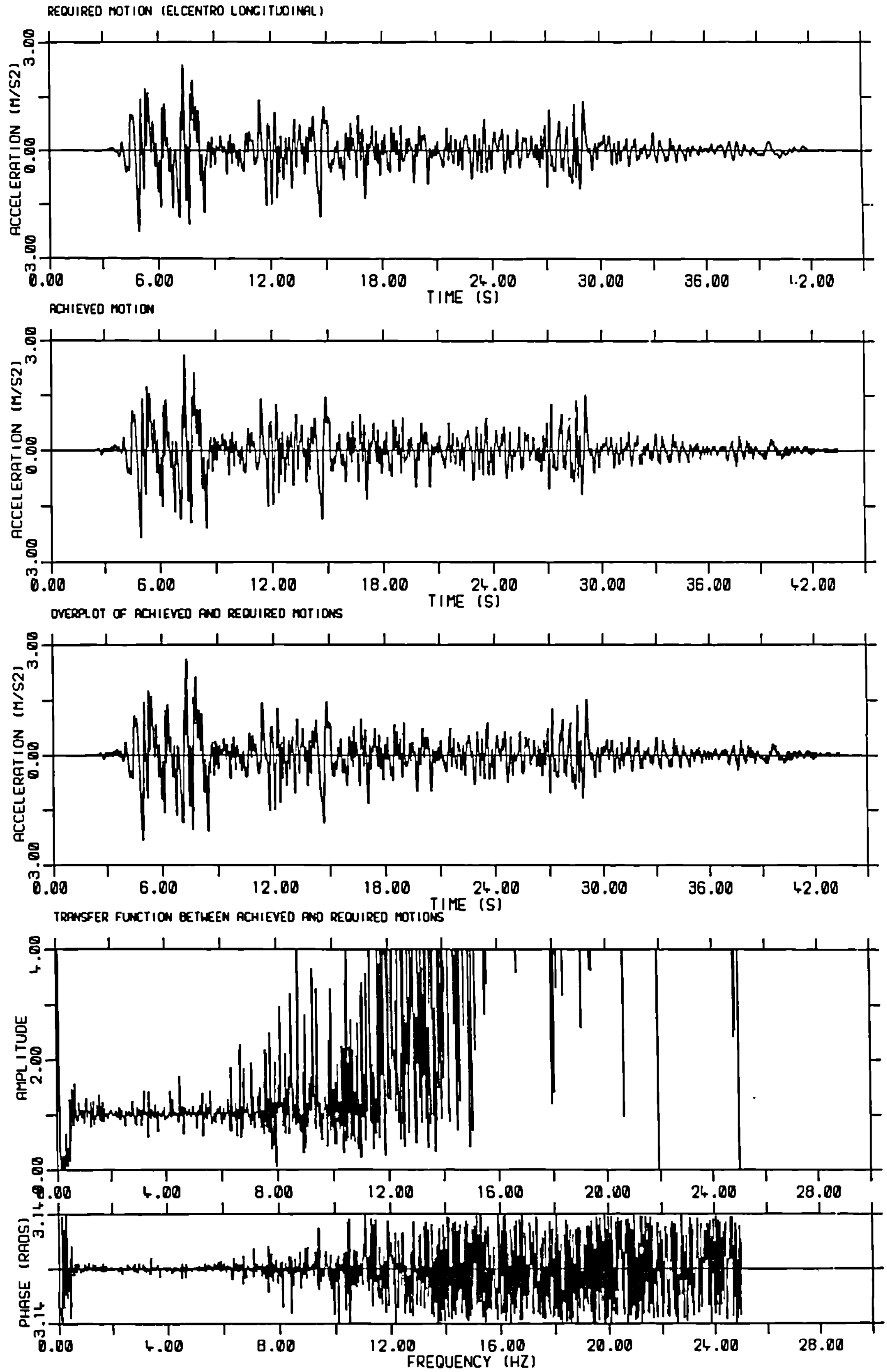


Fig. 5.12 Acceleration time history achieved on the Bristol table for an acceleration match of the El Centro shake with no payload

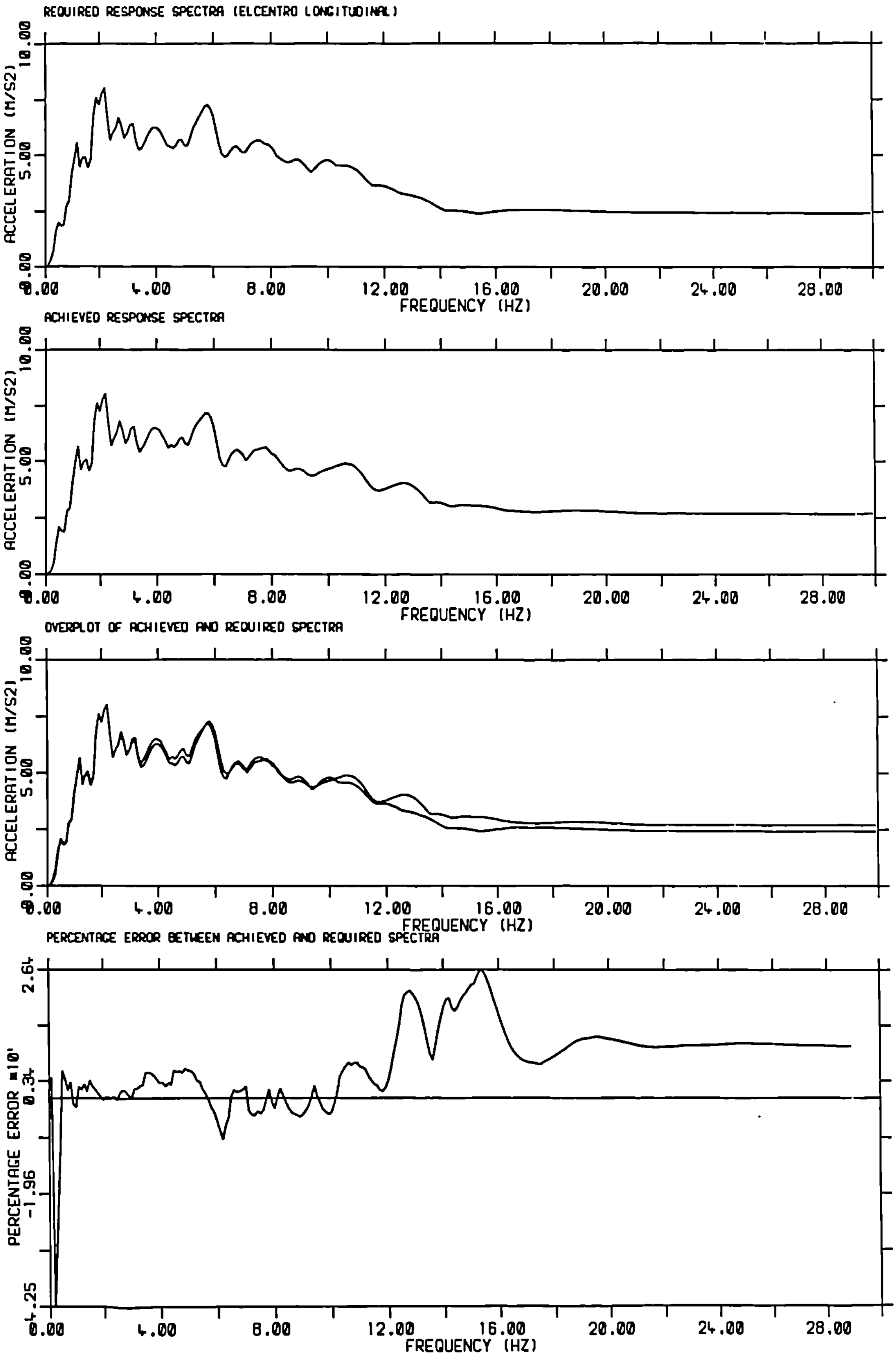


Fig. 5.13 Response spectrum achieved on the Bristol table for an acceleration match of the El Centro shake with no payload

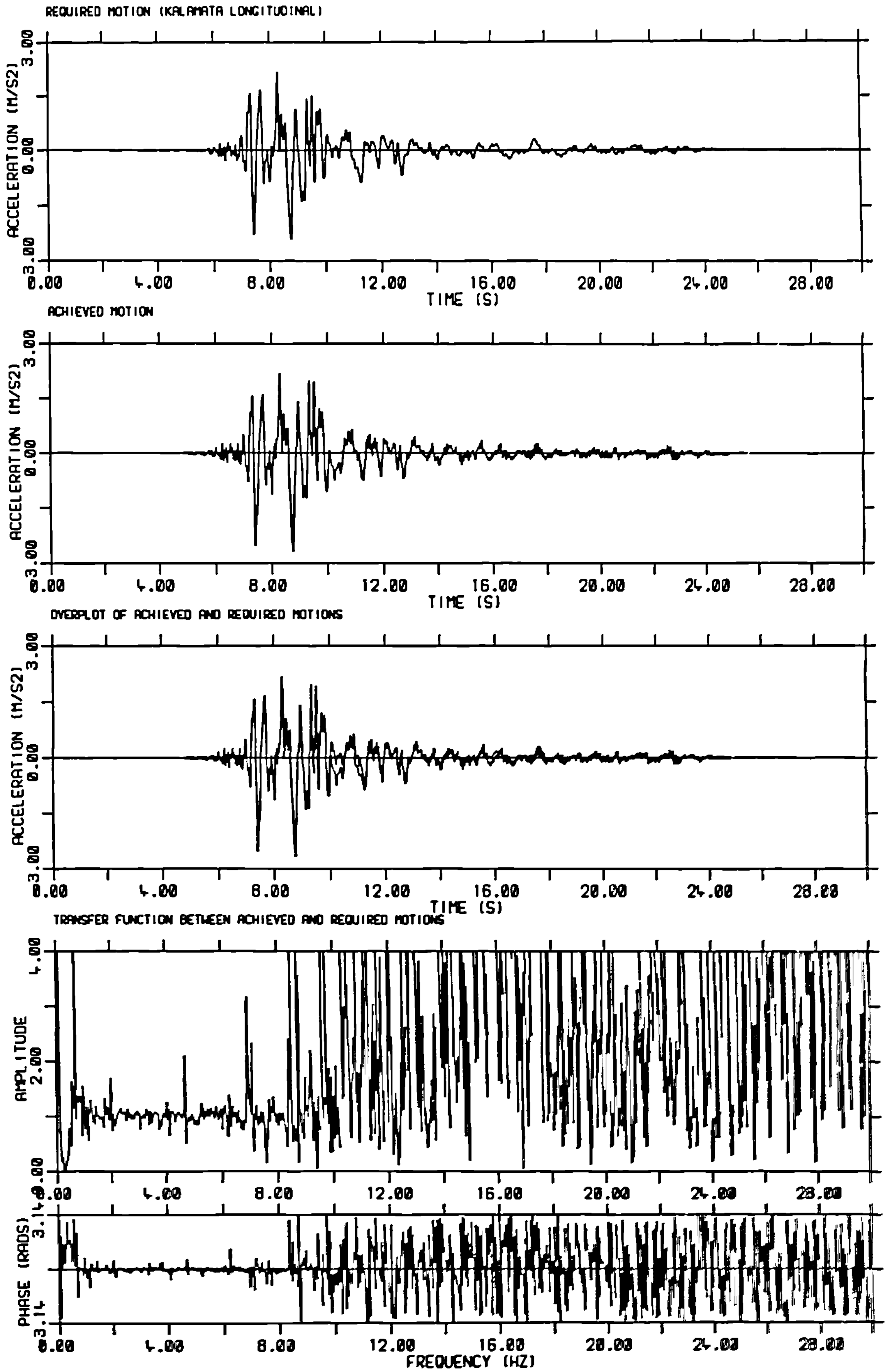


Fig. 5.14 Acceleration time history achieved on the Bristol table for an acceleration match of the Kalamata shake with no payload

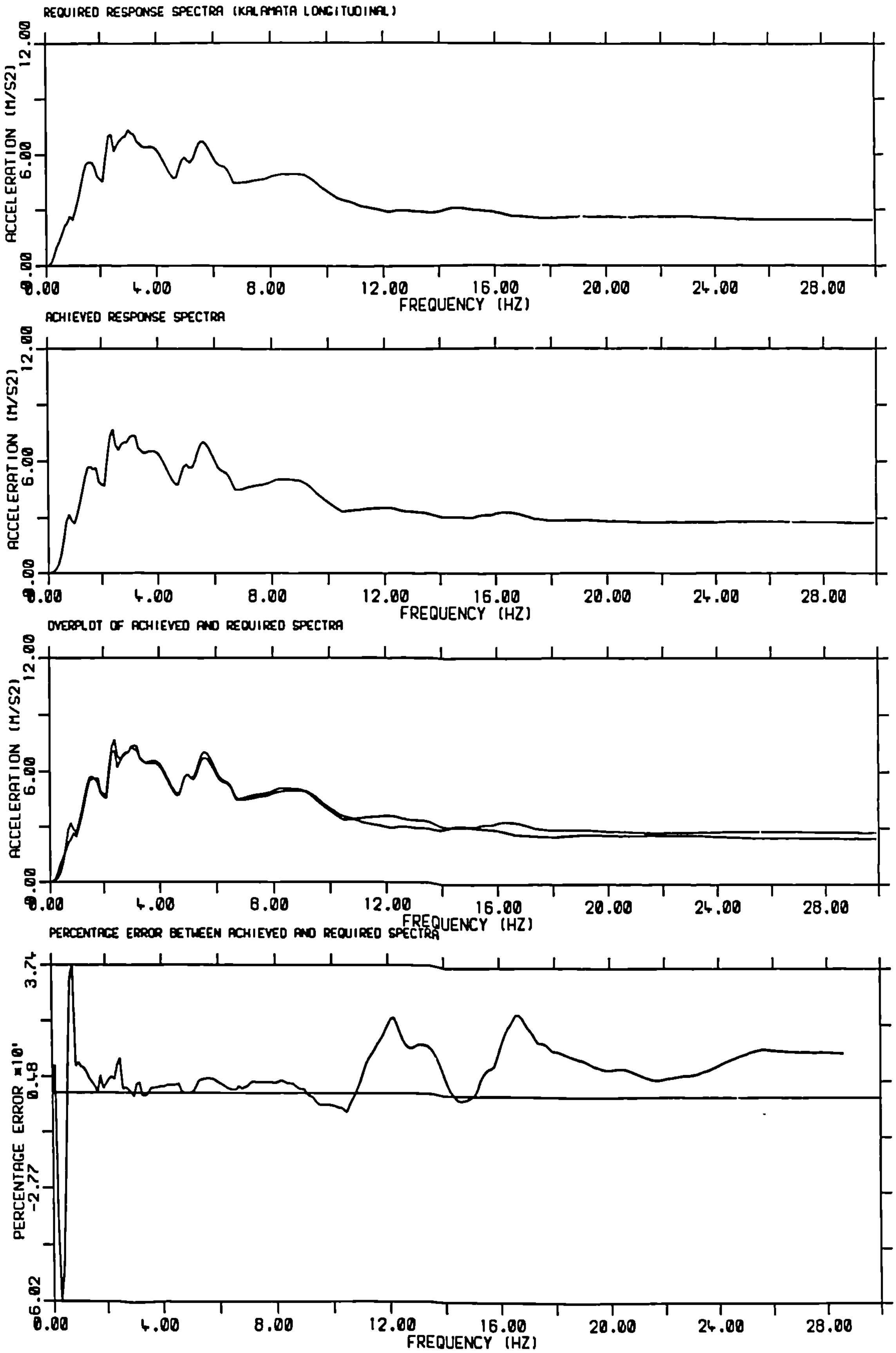


Fig. 5.15 Response spectrum achieved on the Bristol table for an acceleration match of the Kalamata shake with no payload

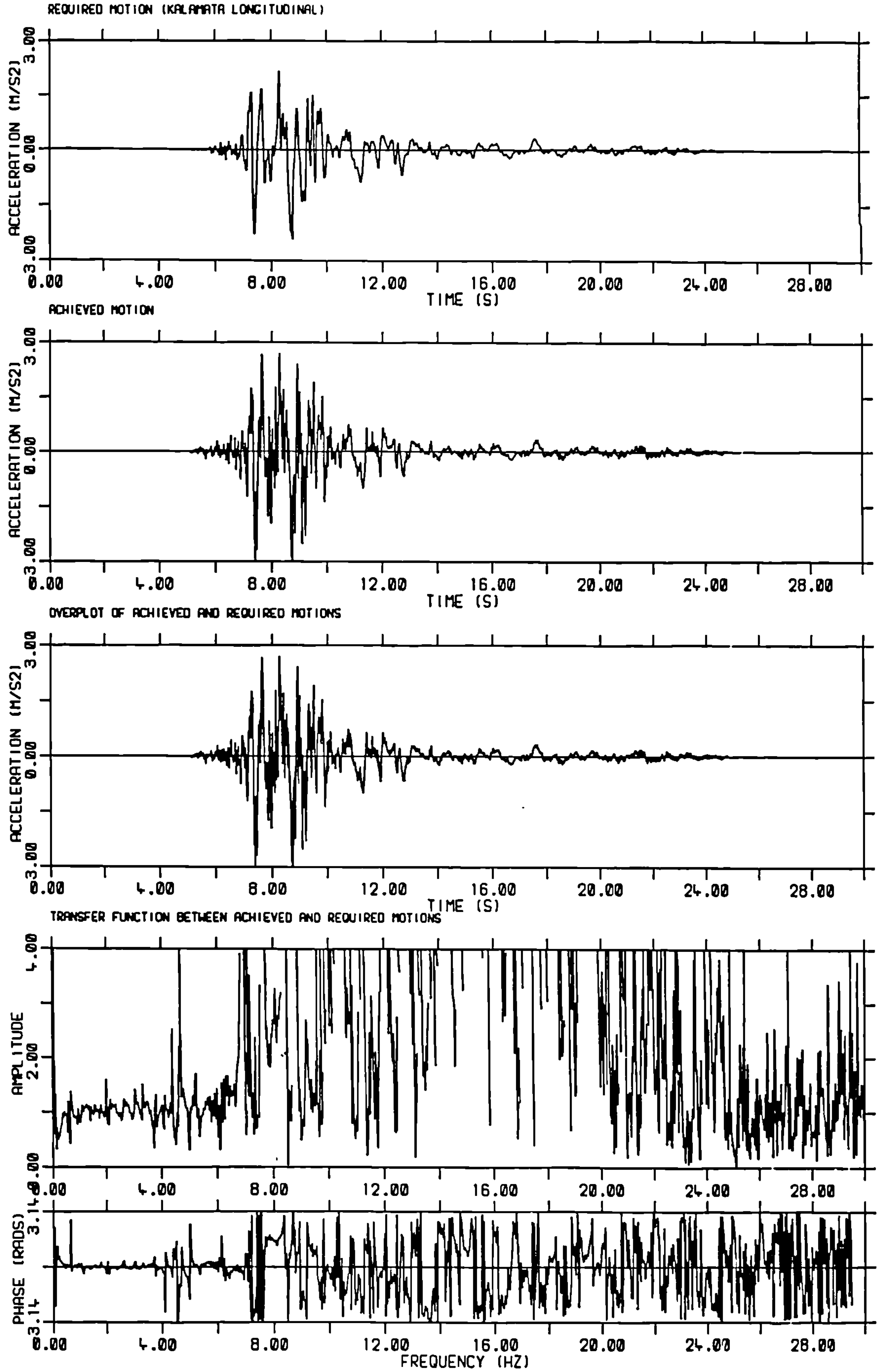


Fig. 5.16 Acceleration time history achieved on the Bristol table for an acceleration match of the Kalamata shake with the 5 tonne flexible specimen

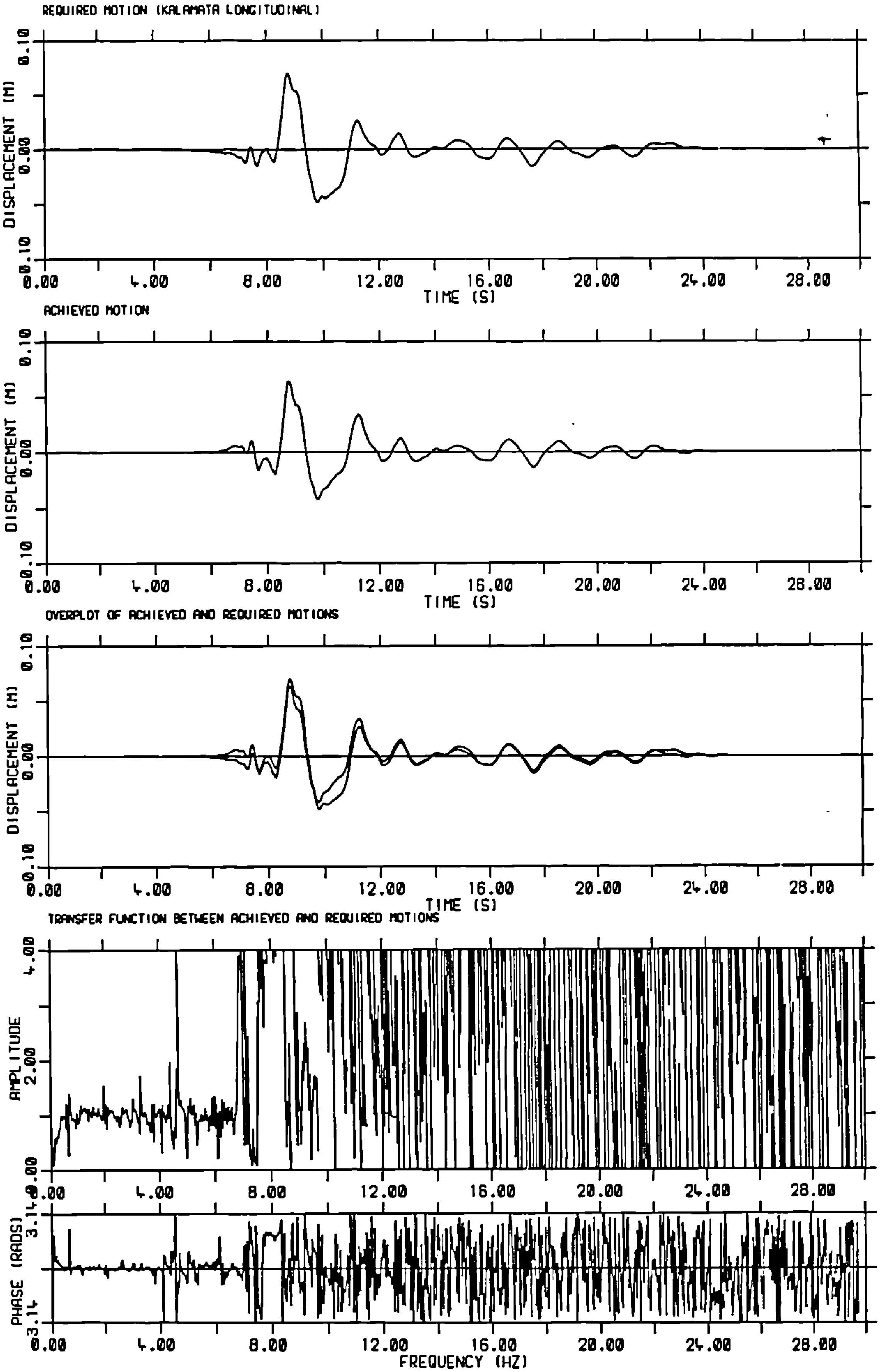


Fig. 5.17 Displacement time history achieved on the Bristol table for an acceleration match of the Kalamata shake with the 5 tonne flexible specimen

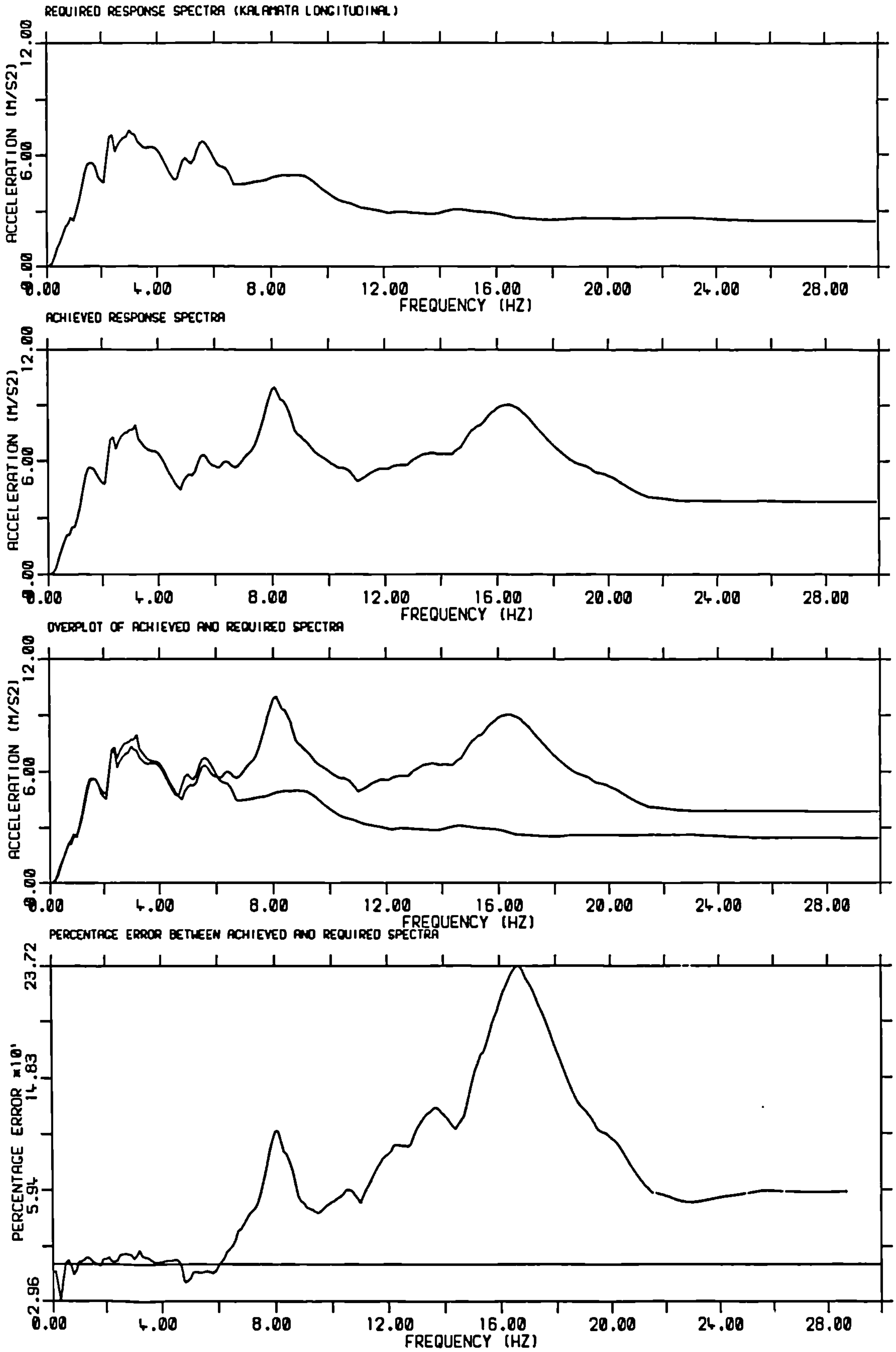


Fig. 5.18 Response spectrum achieved on the Bristol table for an acceleration match of the Kalamata shake with the 5 tonne flexible specimen

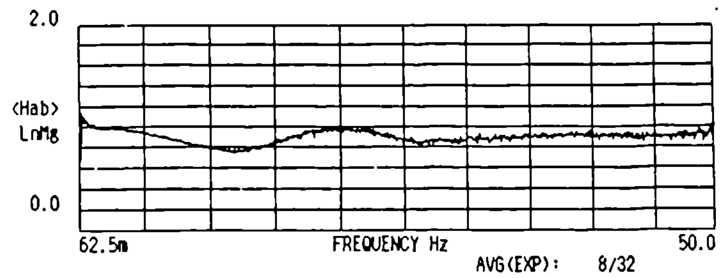


Fig. 5.19 Typical frequency response function of the Athens table with no payload after tuning

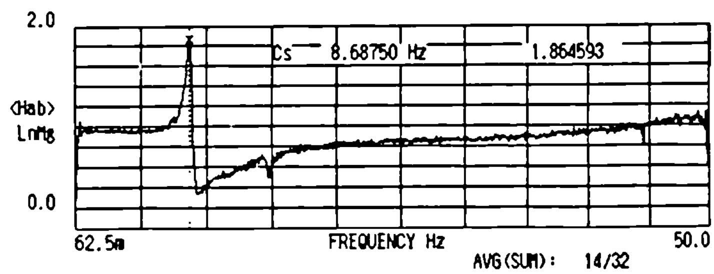


Fig. 5.20 Typical frequency response function of the Athens table after the flexible payload has been added but before the system is re-tuned

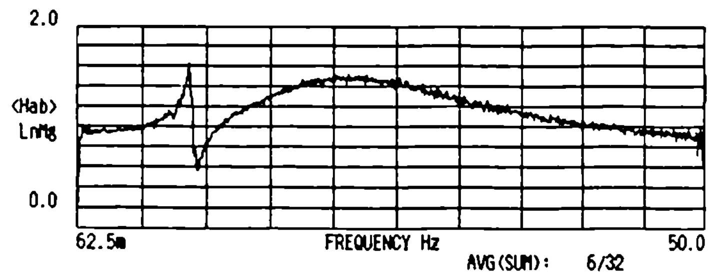


Fig. 5.21 Typical frequency response function of the Athens table after the flexible payload has been added and after the system has been re-tuned

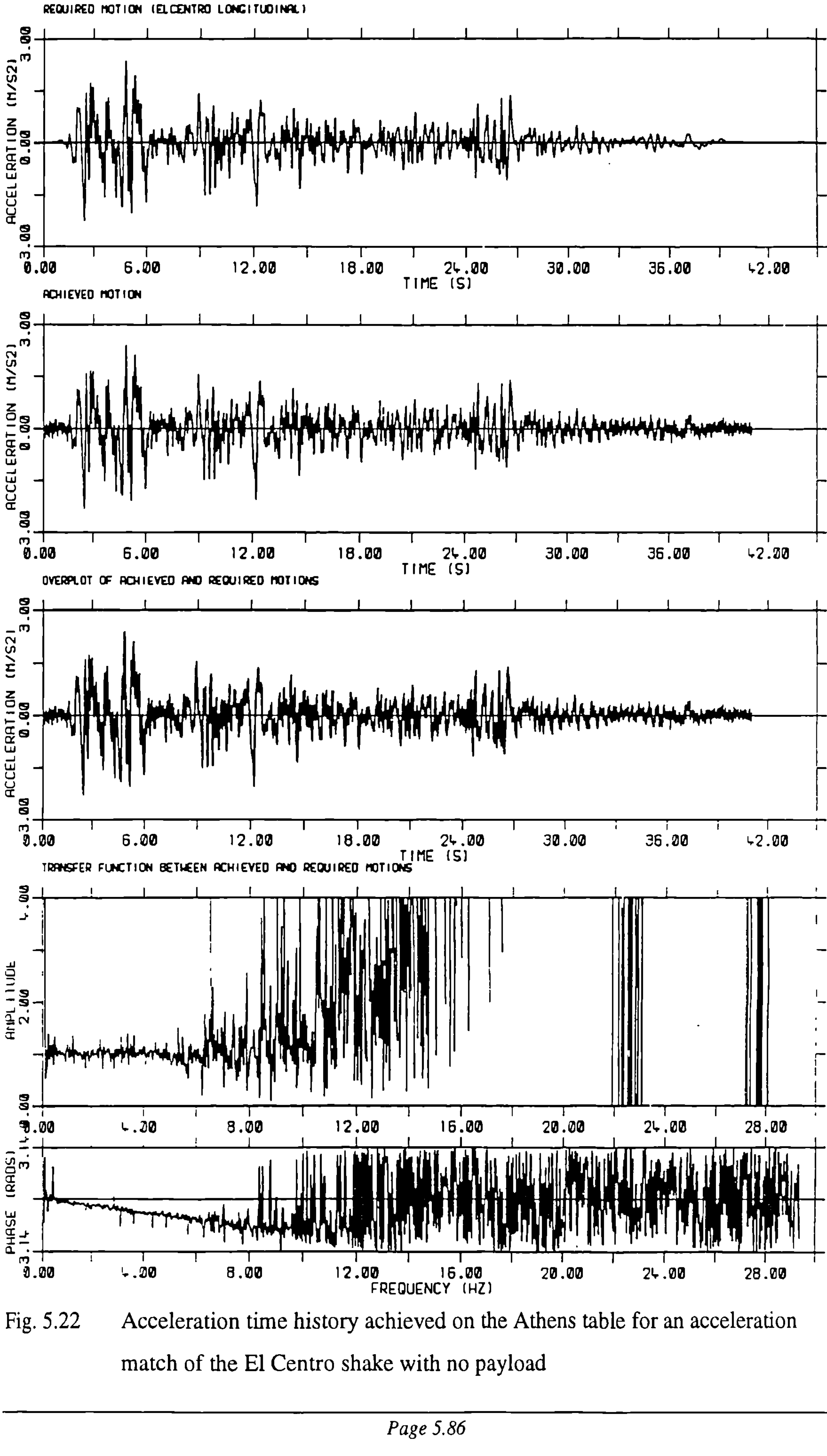


Fig. 5.22 Acceleration time history achieved on the Athens table for an acceleration match of the El Centro shake with no payload

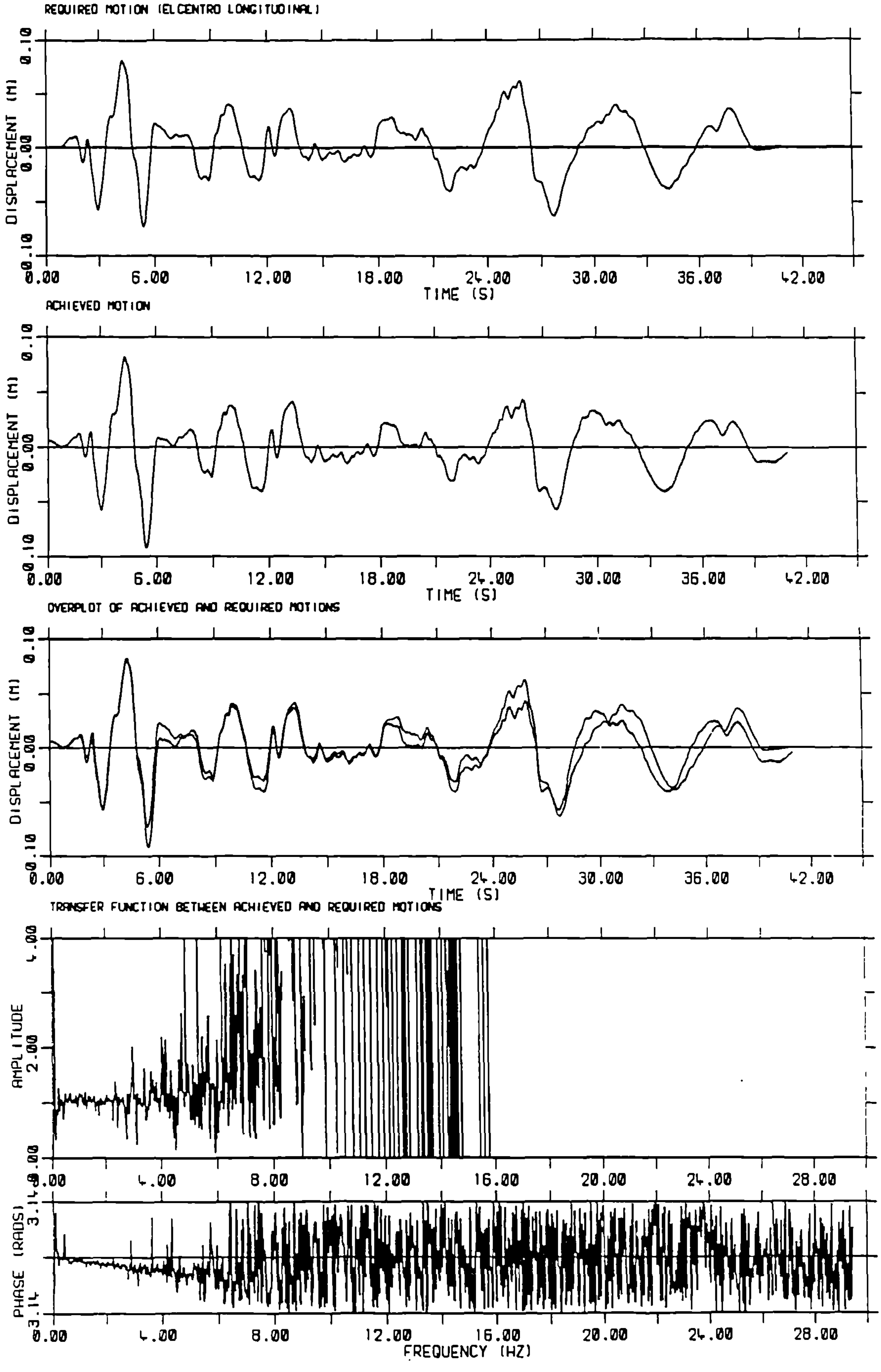


Fig. 5.23 Displacement time history achieved on the Athens table for an acceleration match of the El Centro shake with no payload

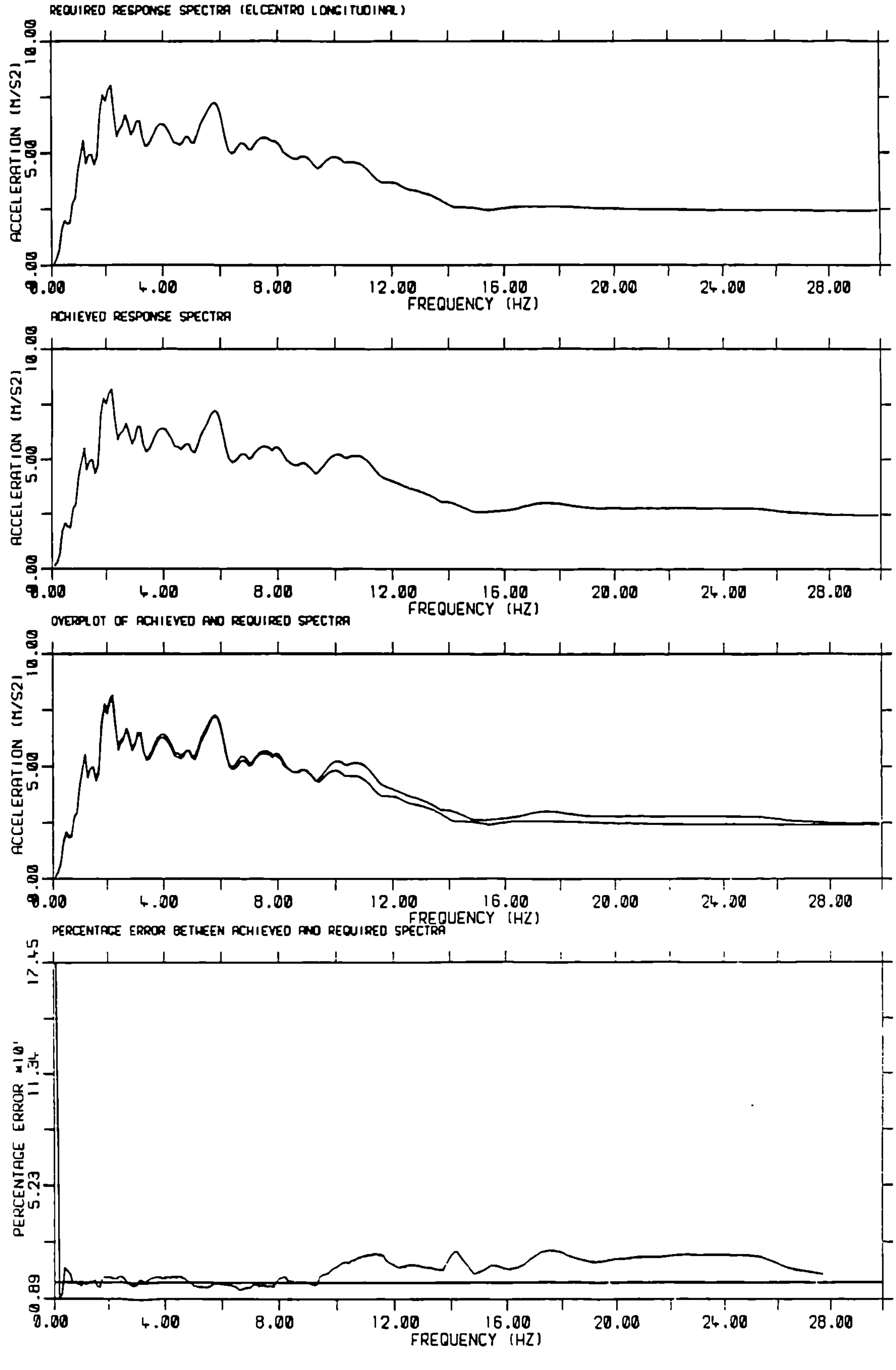


Fig. 5.24 Response spectrum achieved on the Athens table for an acceleration match of the El Centro shake with no payload

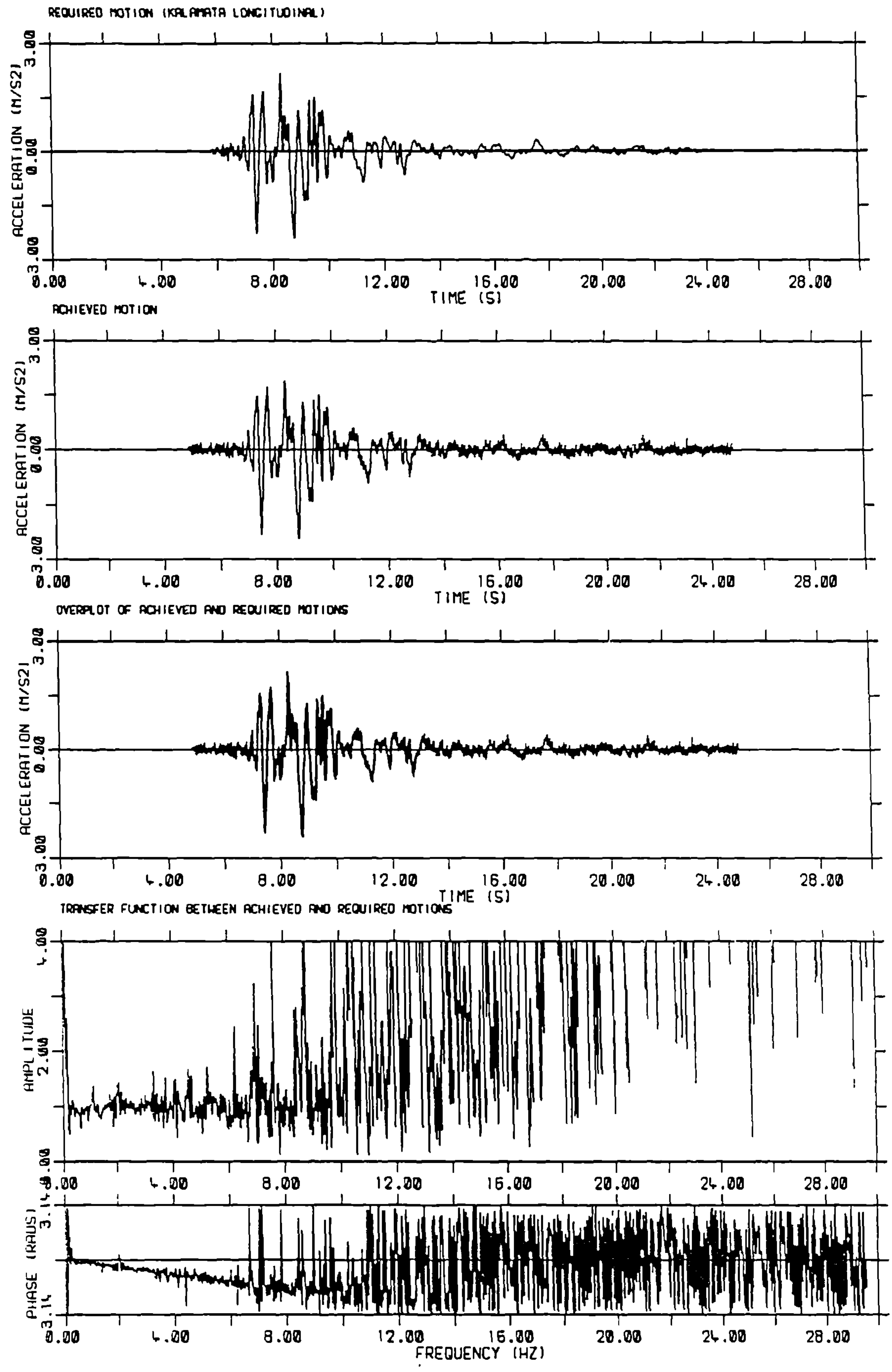


Fig. 5.25 Acceleration time history achieved on the Athens table for an acceleration match of the Kalamata shake with no payload

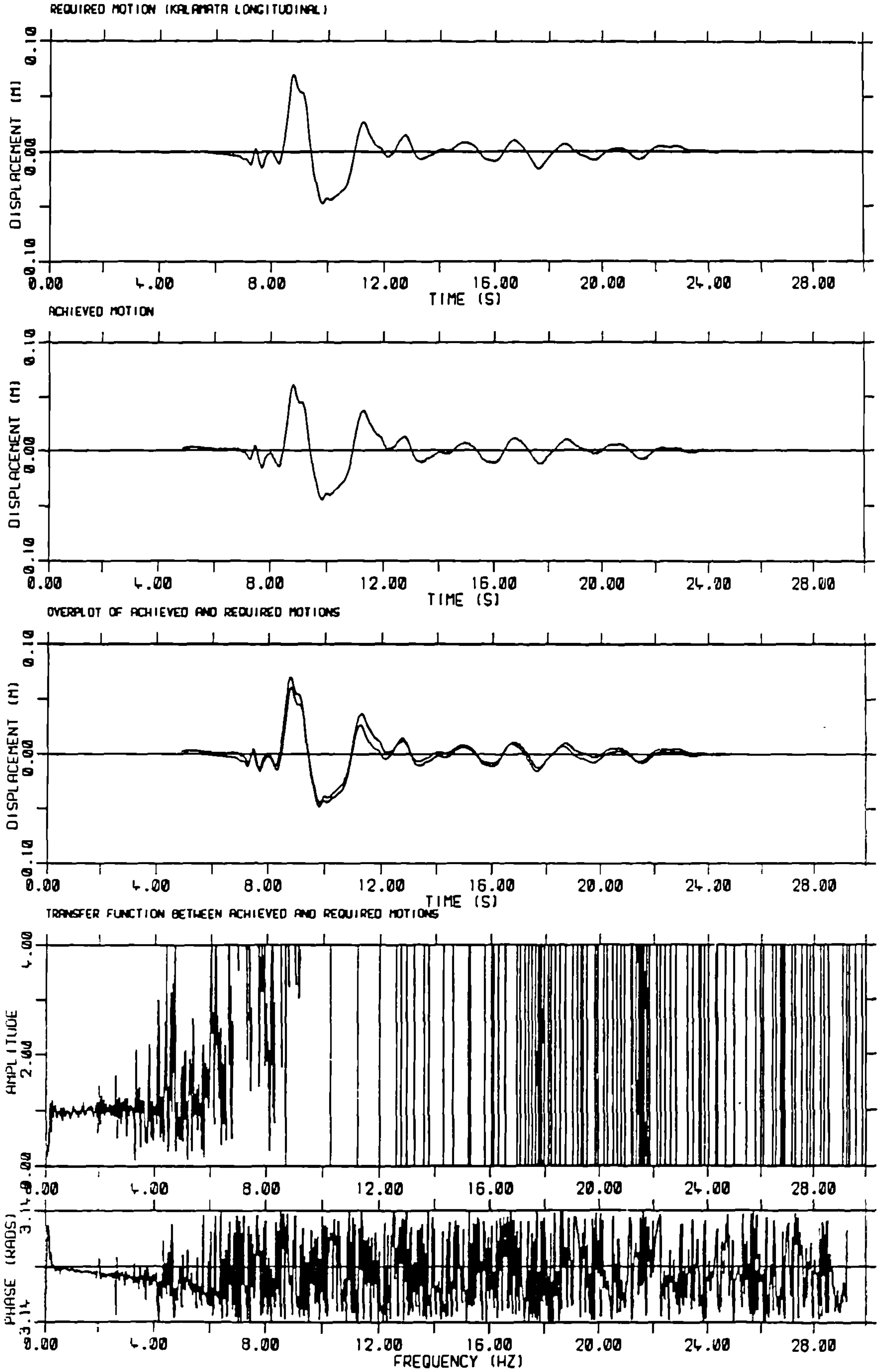


Fig. 5.26 Displacement time history achieved on the Athens table for an acceleration match of the Kalamata shake with no payload

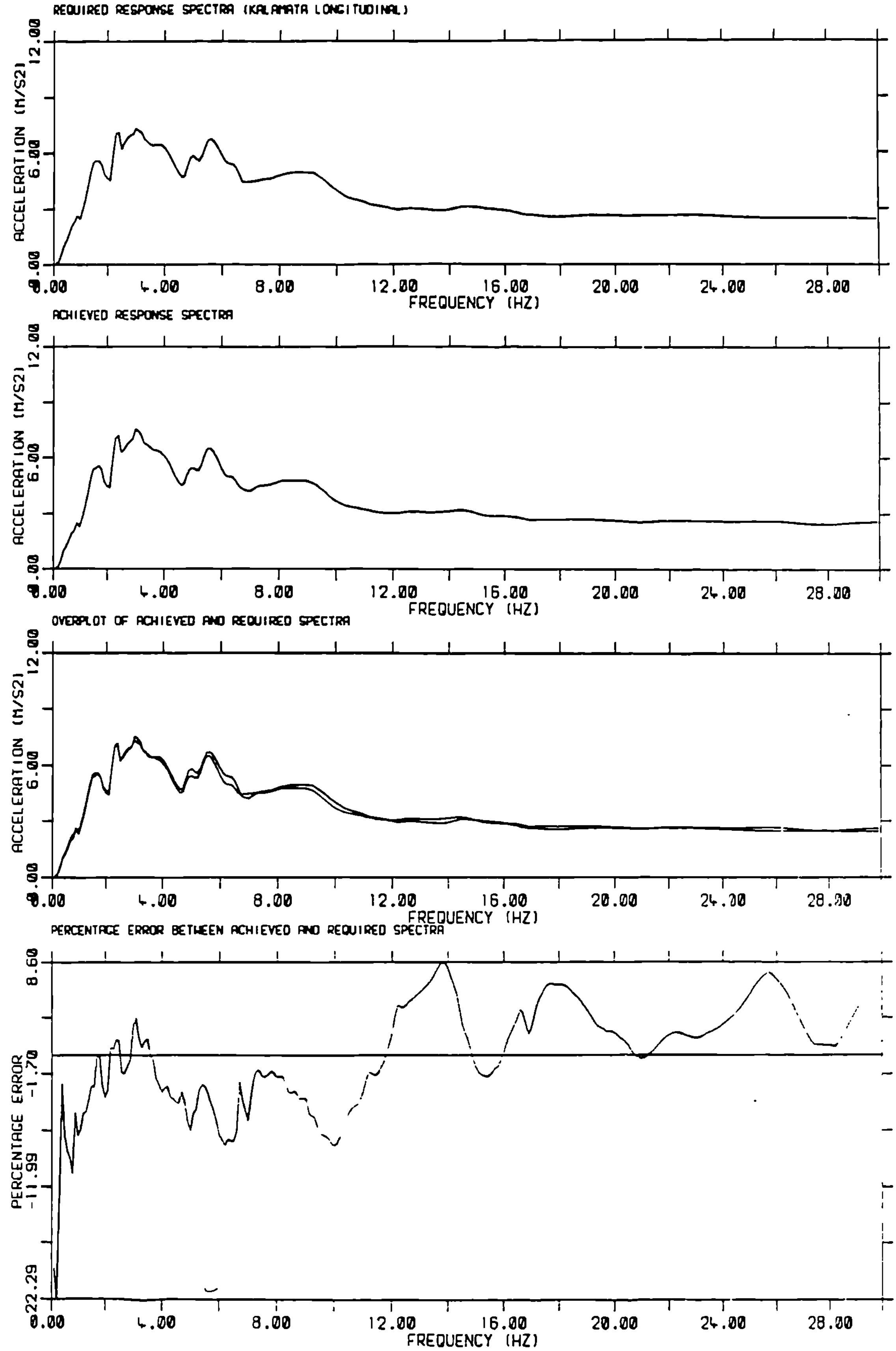


Fig. 5.27 Response spectrum achieved on the Athens table for an acceleration match of the Kalamata shake with no payload

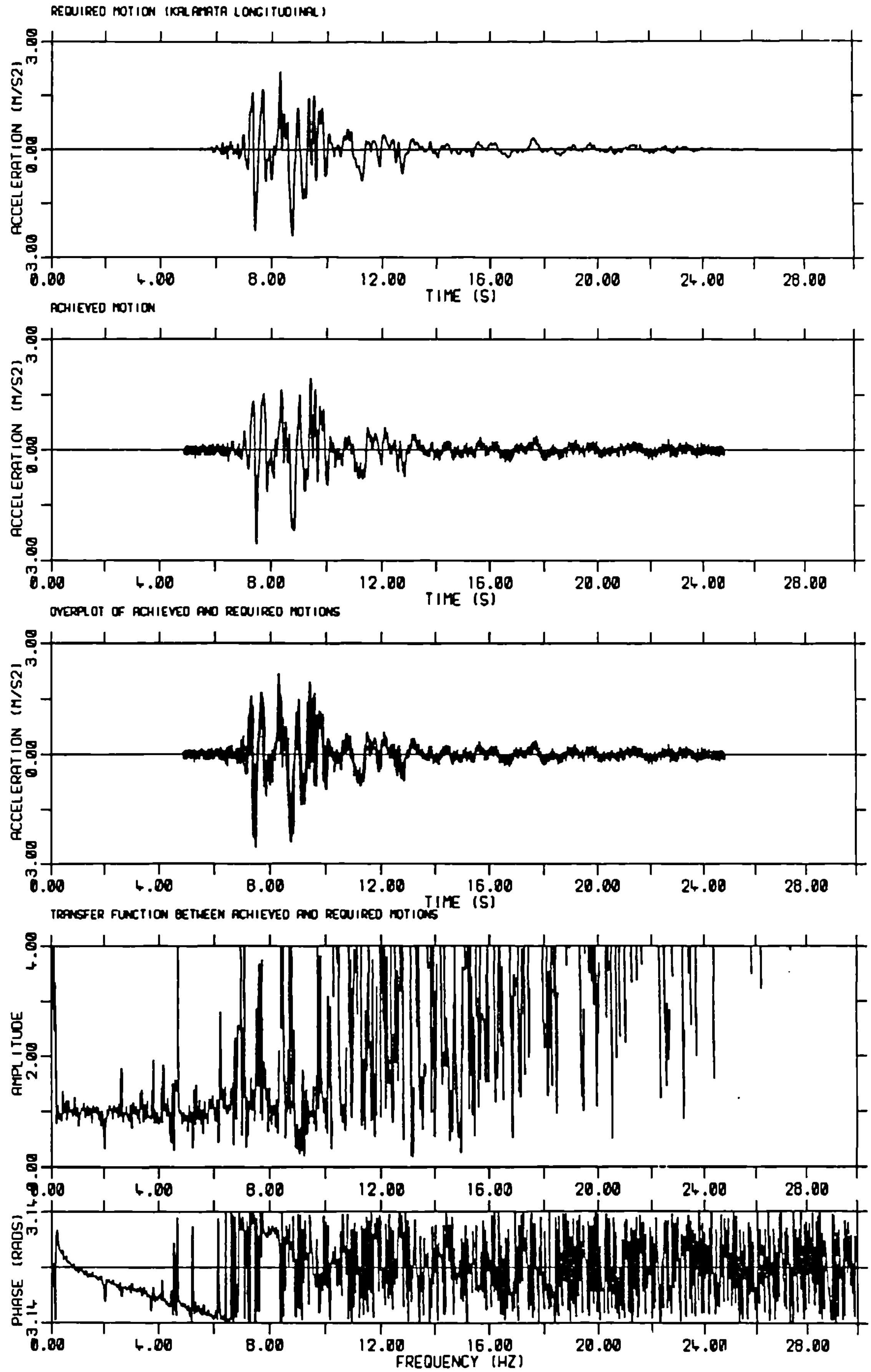


Fig. 5.28 Acceleration time history achieved on the Athens table for an acceleration match of the Kalamata shake with the 5 tonne flexible specimen

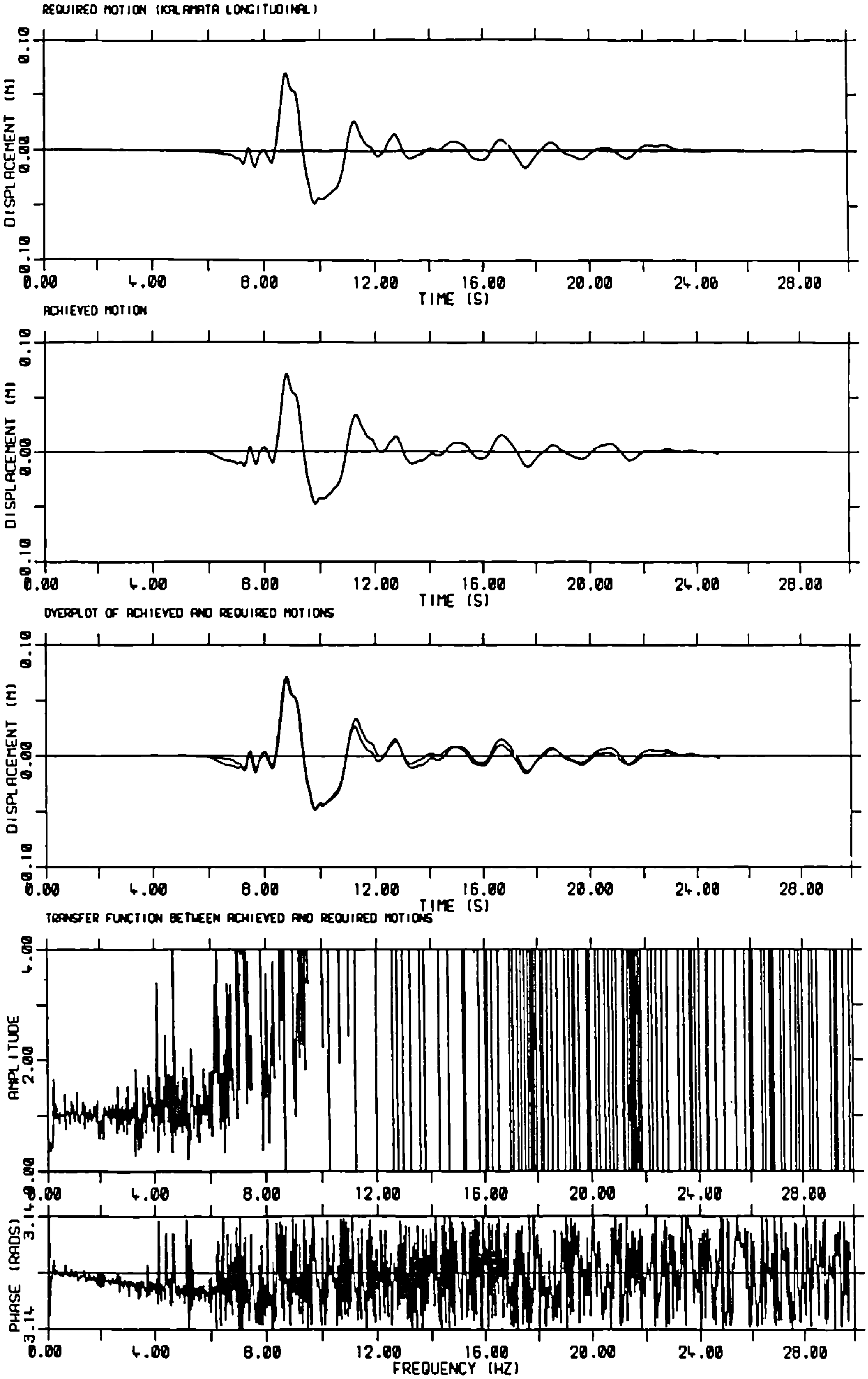


Fig. 5.29 Displacement time history achieved on the Athens table for an acceleration match of the Kalamata shake with the 5 tonne flexible specimen

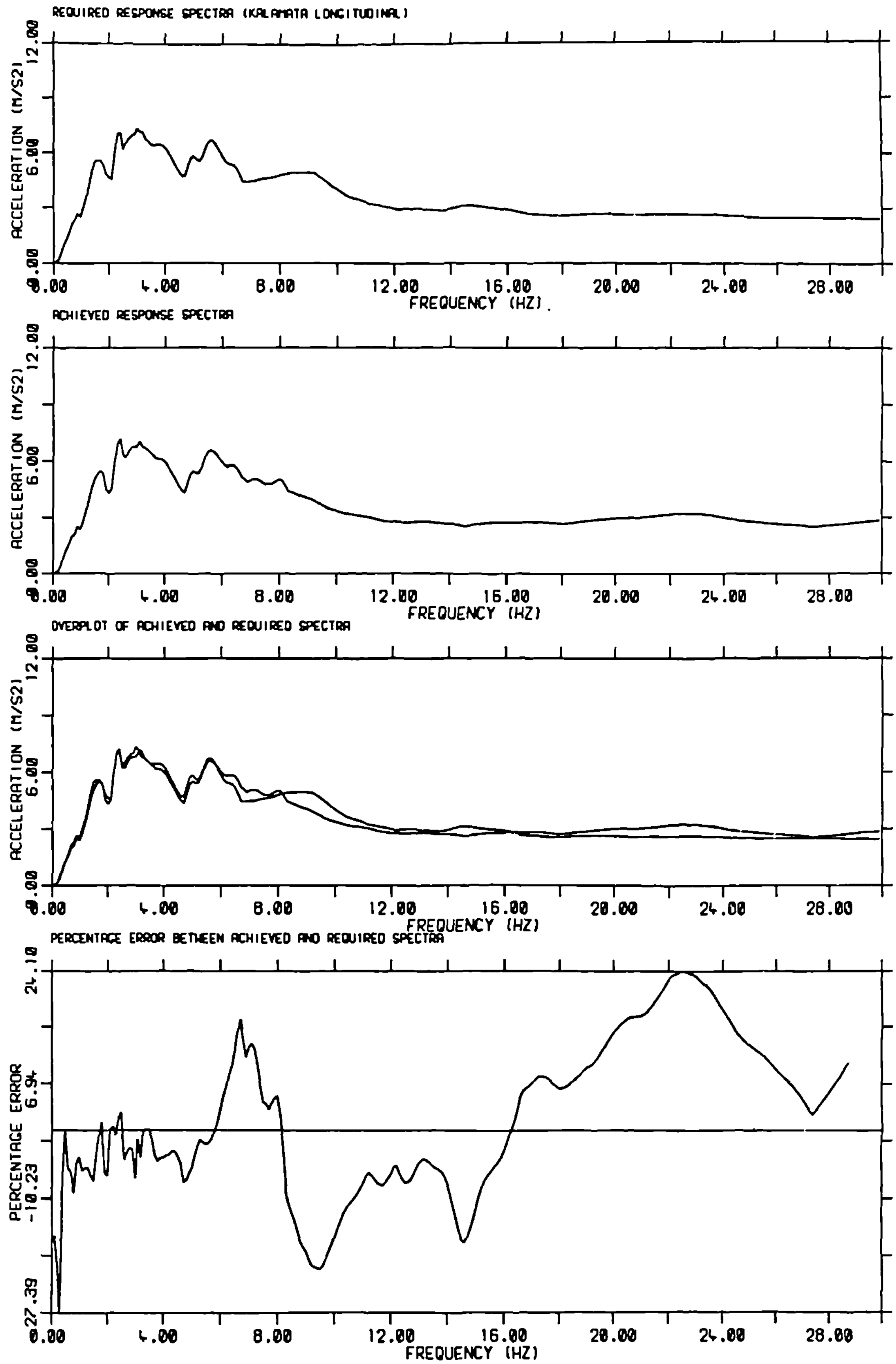


Fig. 5.30 Response spectrum achieved on the Athens table for an acceleration match of the Kalamata shake with the 5 tonne flexible specimen

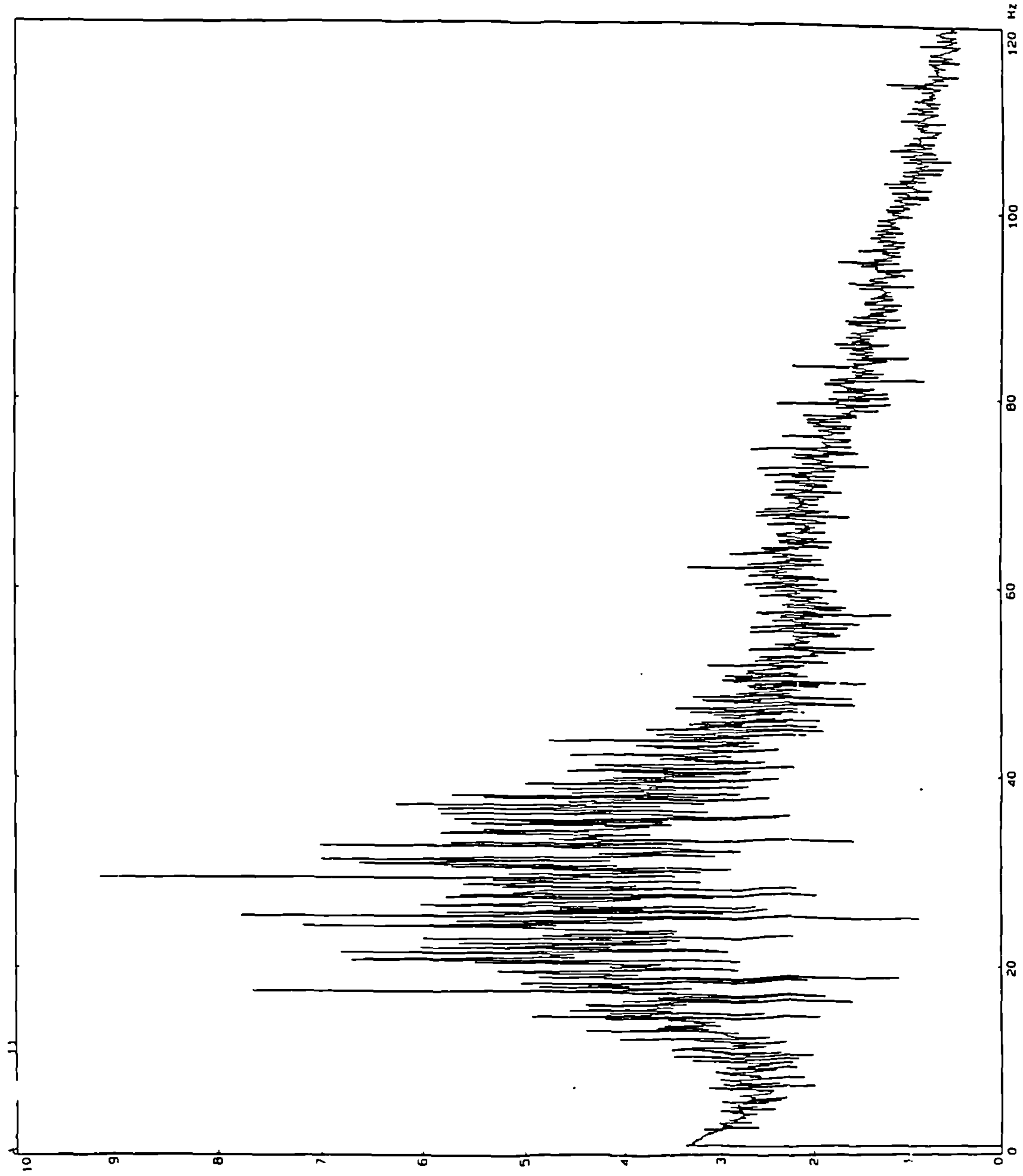


Fig. 5.31 Typical frequency response function of the ISMES table with no payload after tuning

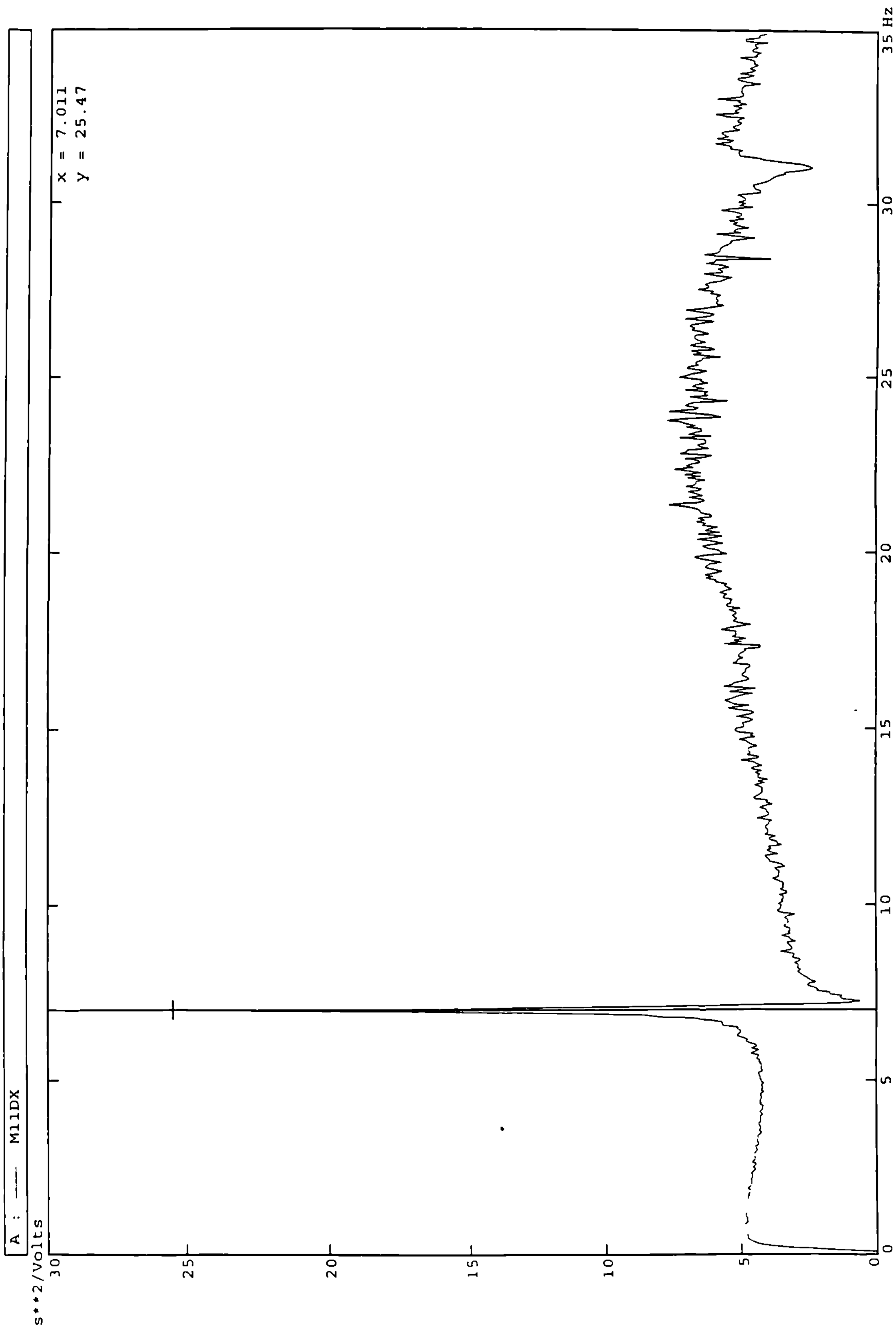


Fig. 5.32 Typical frequency response function of the ISMES table after the flexible payload has been added but before the system is re-tuned

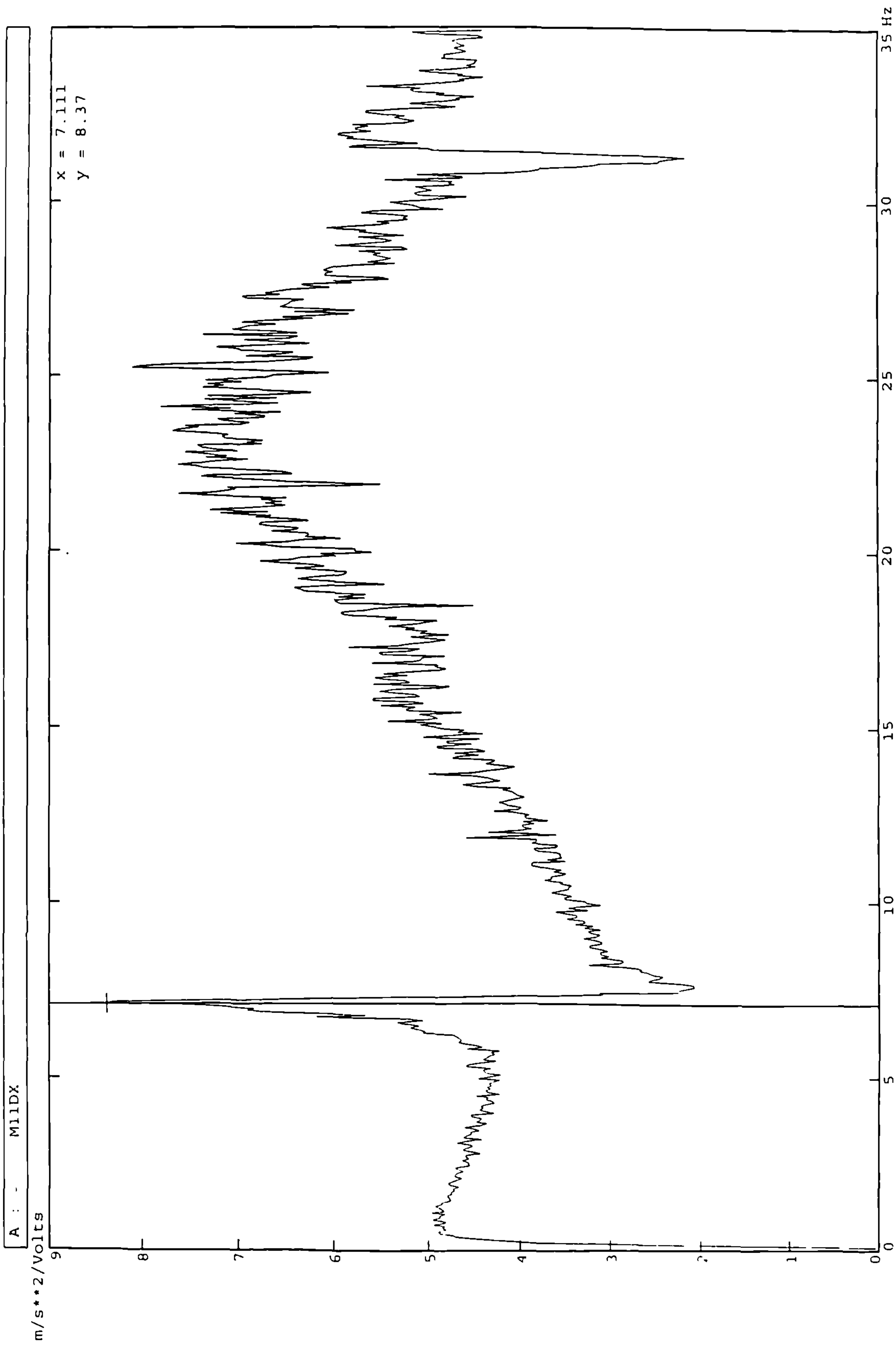
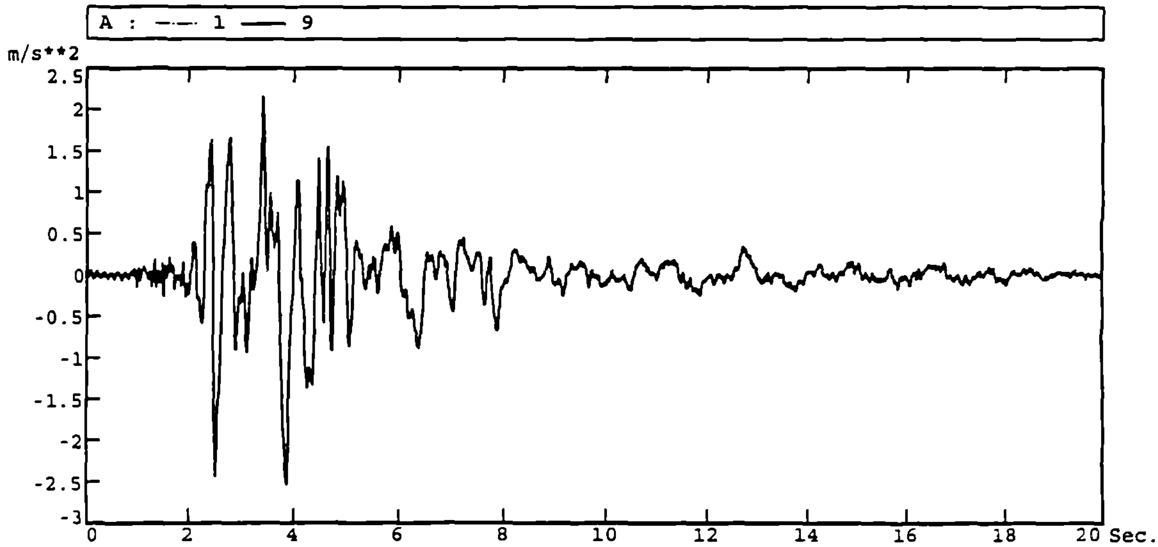


Fig. 5.33 Typical frequency response function of the ISMES table after the flexible payload has been added and after the system has been re-tuned

Table conditions : with flexible payload 8t
Controlled DOF : 6
Control mode : acceleration
Drive filter = 0.2 - 35 Hz

Test name : istar
Version name : flex6
save n.18
Level = 0 dB

Overplot of achieved and required acceleration (dotted line) in X axis



Overplot of achieved and required displacement (dotted line) in X axis

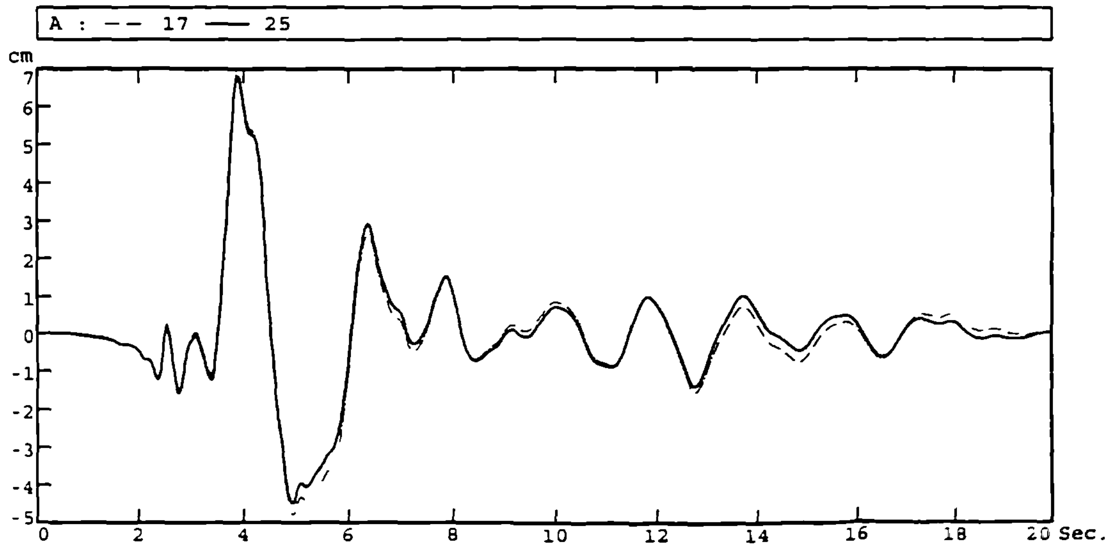
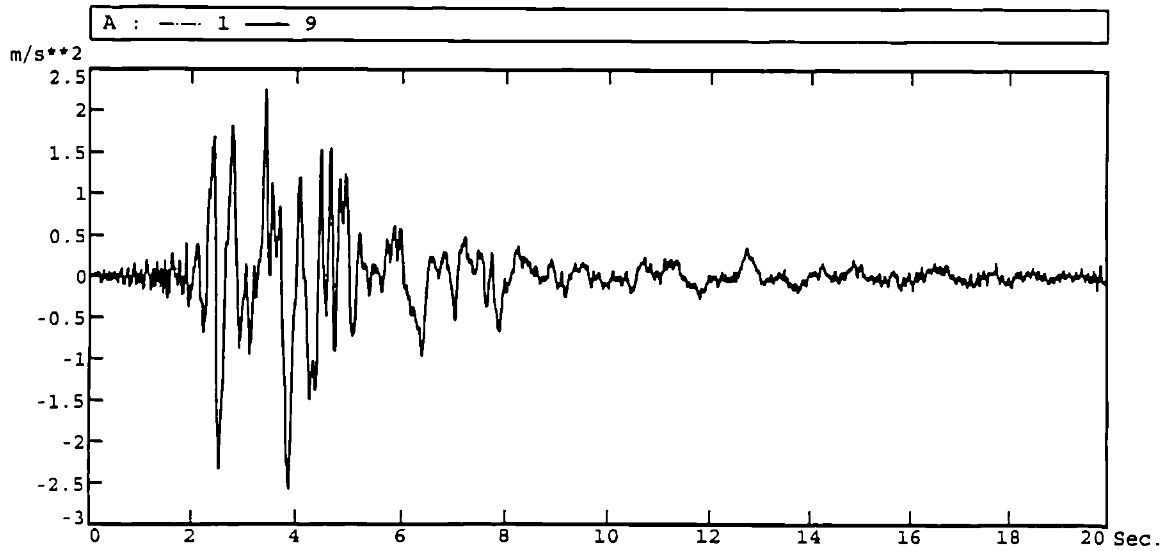


Fig. 5.34 Time histories achieved on the ISMES table for a 6 DOF acceleration match of the Kalamata shake with the 8 tonne flexible specimen

Table conditions : with flexible payload 8t
Controlled DOF : 6
Control mode : displacement
Drive filter = 0.1 - 20 Hz

Test name : ecoest
Version name : flex6d
save n.5
Level = 0 dB

Overplot of achieved and required acceleration (dotted line) in X axis



Overplot of achieved and required displacement (dotted line) in X axis

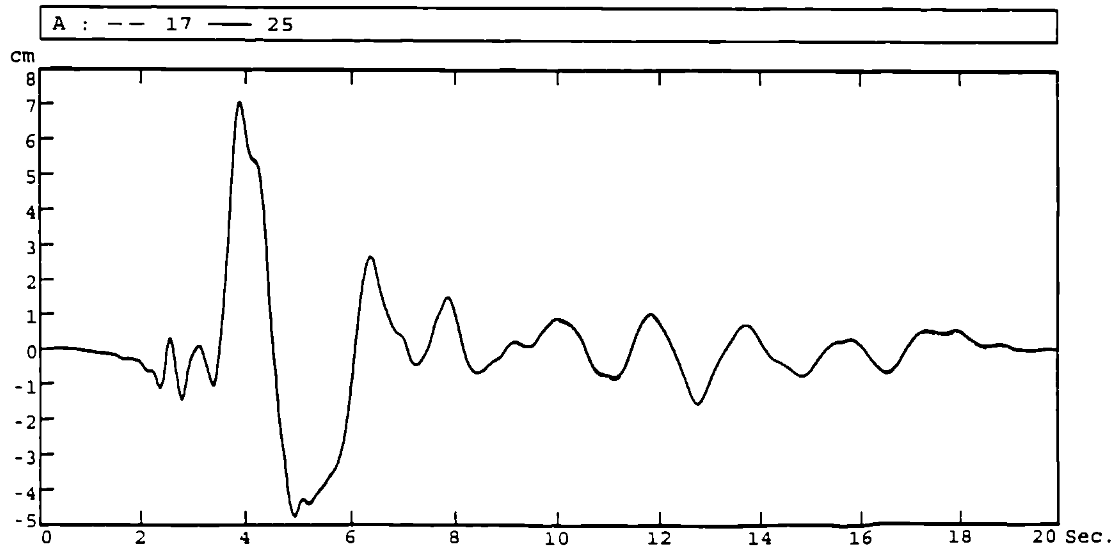
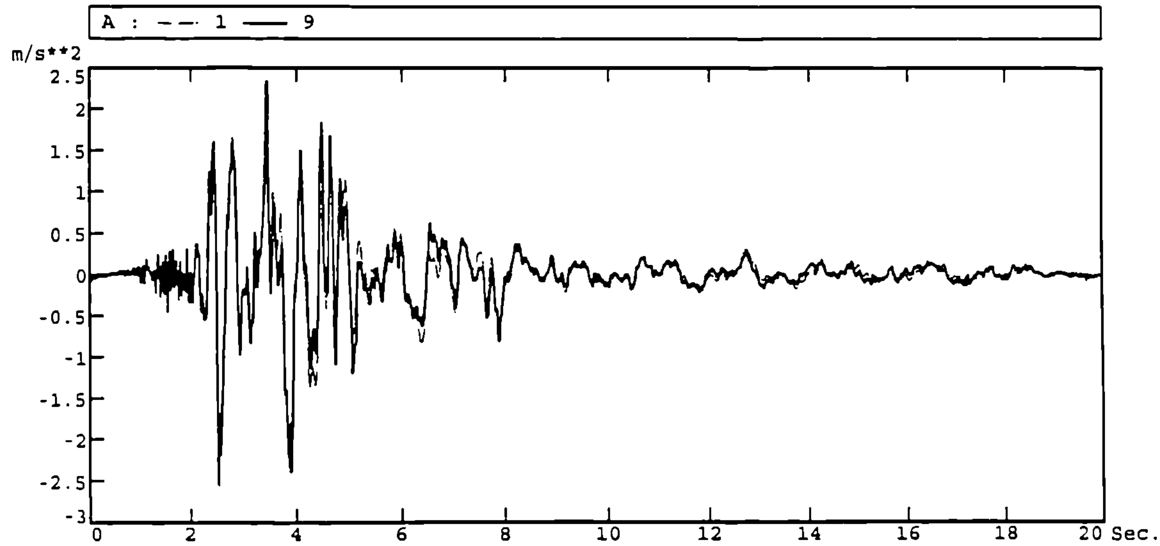


Fig. 5.35 Time histories achieved on the ISMES table for a 6 DOF displacement match of the Kalamata shake with the 8 tonne flexible specimen

Table conditions : with rigid payload 8t
Controlled DOF : 0
Control mode : acceleration
Drive filter = 0.5 - 35 Hz

Test name : ecoest
Version name : kala8t0
save n.1
Level = 0 dB

Overplot of achieved and required acceleration (dotted line) in X axis



Overplot of achieved and required displacement (dotted line) in X axis

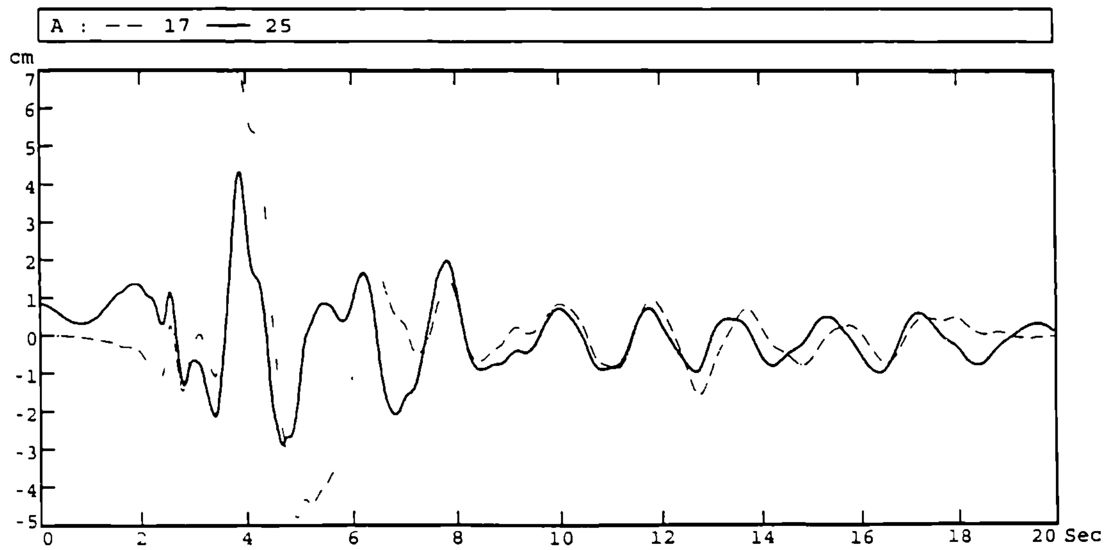
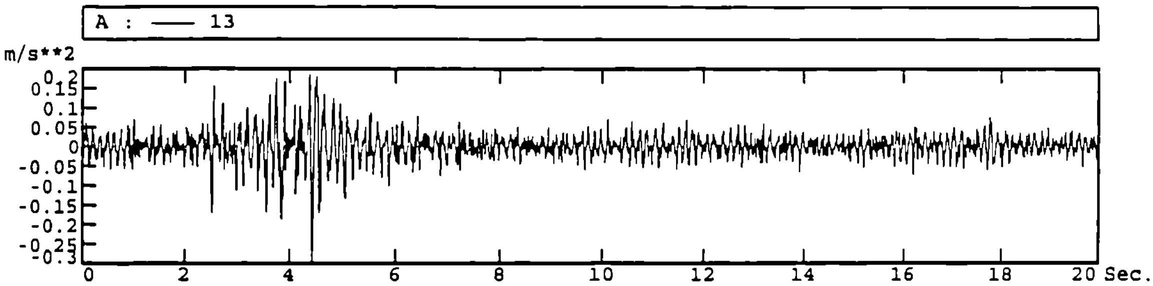


Fig. 5.36 Time histories achieved on the ISMES table with no software compensation of the Kalamata shake with the 8 tonne rigid payload

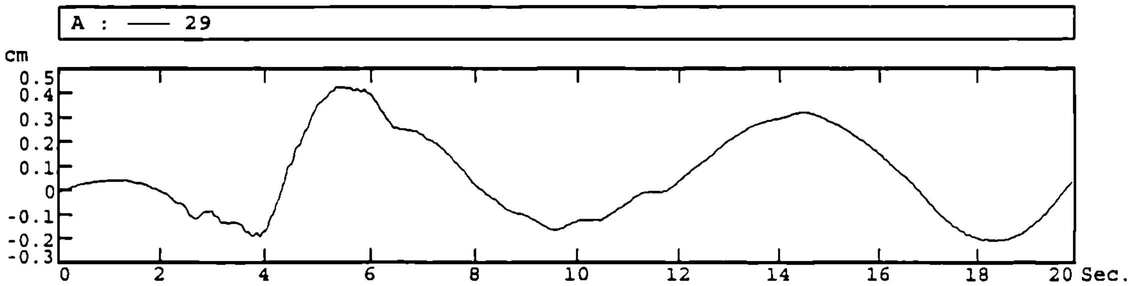
Table conditions : with flexible payload 8t
Controlled DOF : 6
Control mode : acceleration
Drive filter = 0.2 - 35 Hz

Test name : istar
Version name : flex6
save n.18
Level = 0 dB

Difference between achieved and required acceleration in X axis



Difference between achieved and required displacement in X axis



Transfer function between achieved and required acceleration in X axis

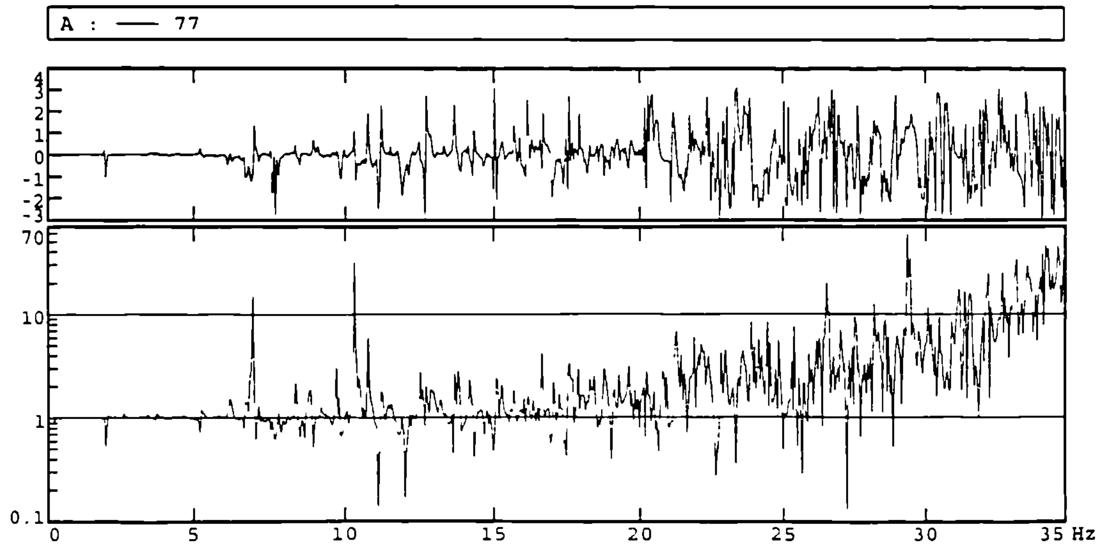
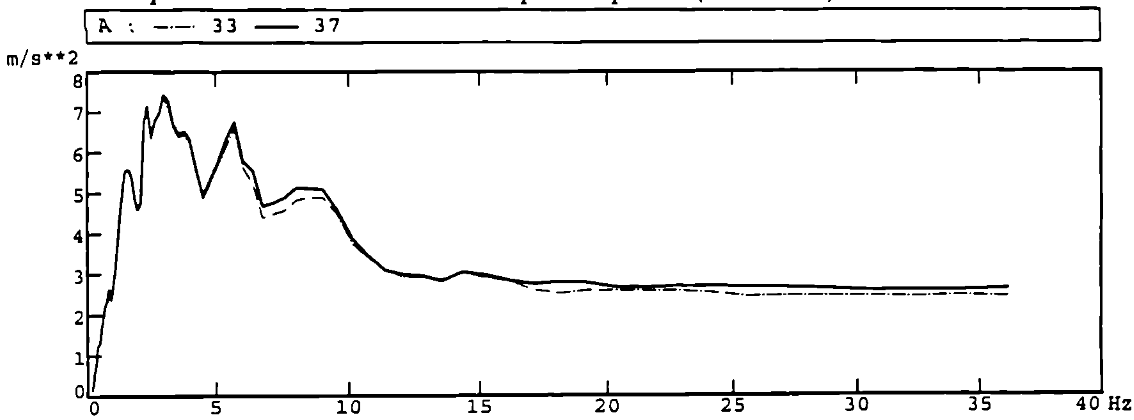


Fig. 5.37 Errors in the acceleration time history matching on the ISMES table for a 6 DOF acceleration match of the Kalamata shake with the 8 tonne flexible specimen

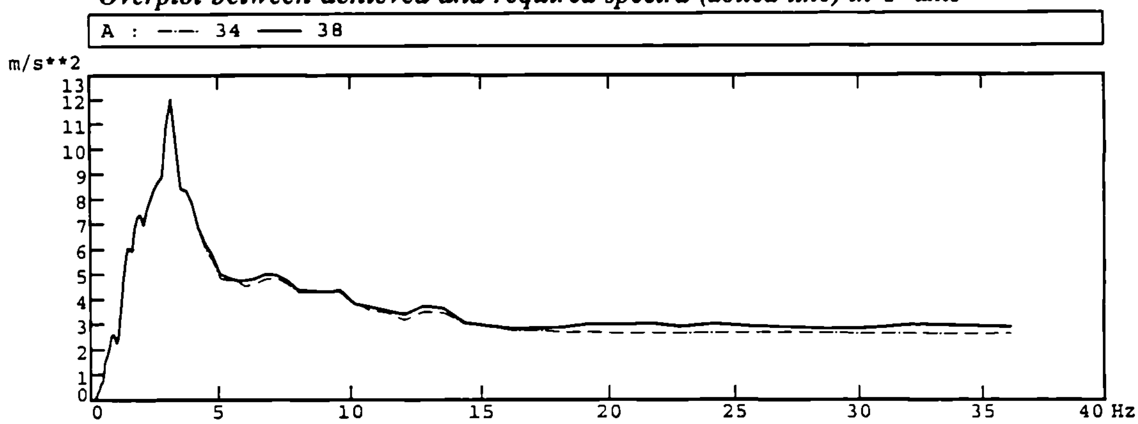
Table conditions : with flexible payload 8t
Controlled DOF : 6
Control mode : acceleration
Drive filter = 0.2 - 35 Hz

Test name : istar
Version name : flex6
save n.18
Level = 0 dB

Overplot between achieved and required spectra (dotted line) in X axis



Overplot between achieved and required spectra (dotted line) in Y axis



Overplot between achieved and required spectra (dotted line) in Z

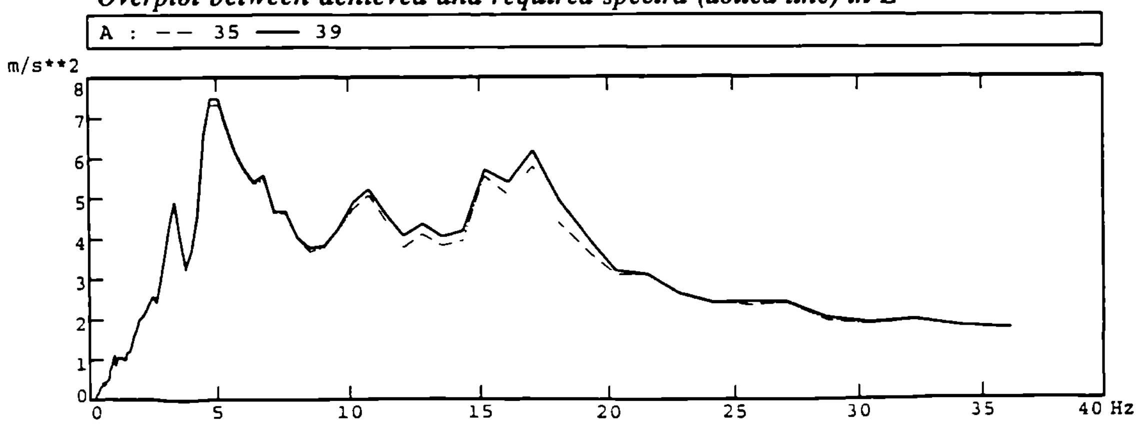
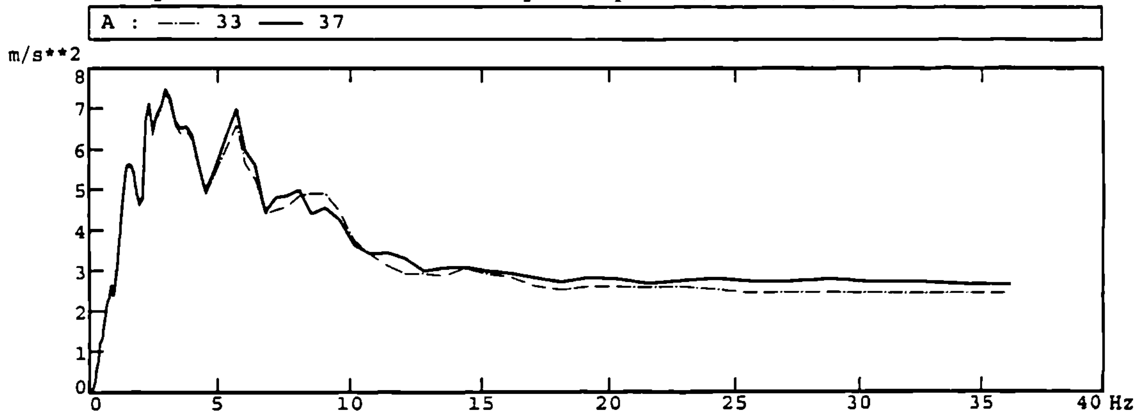


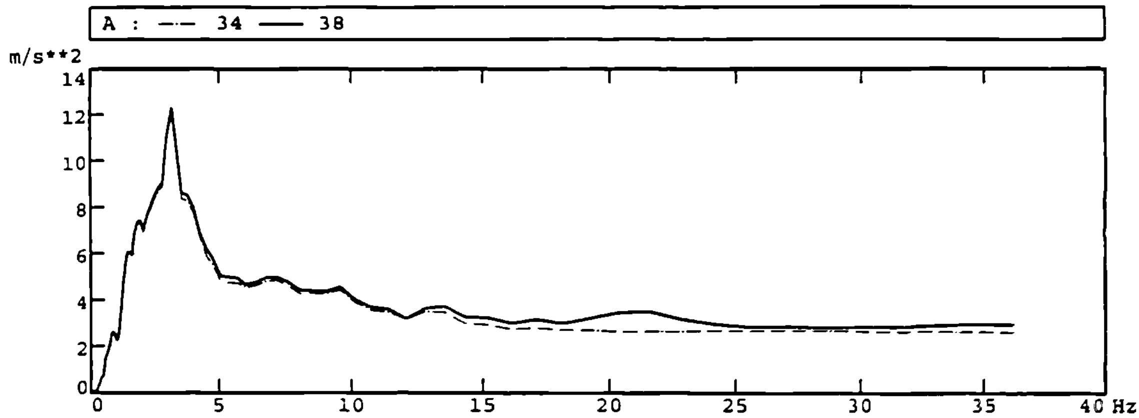
Fig. 5.38 Response spectra achieved on the ISMES table for a 6 DOF acceleration match of the Kalamata shake with the 8 tonne flexible specimen

Table conditions : with flexible payload 8t	Test name : ecoest
Controlled DOF : 6	Version name : flex6d
Control mode : displacement	save n.5
Drive filter = 0.1 - 20 Hz	Level = 0 dB

Overplot between achieved and required spectra (dotted line) in X



Overplot between achieved and required spectra (dotted line) in Y axis



Overplot between achieved and required spectra (dotted line) in Z

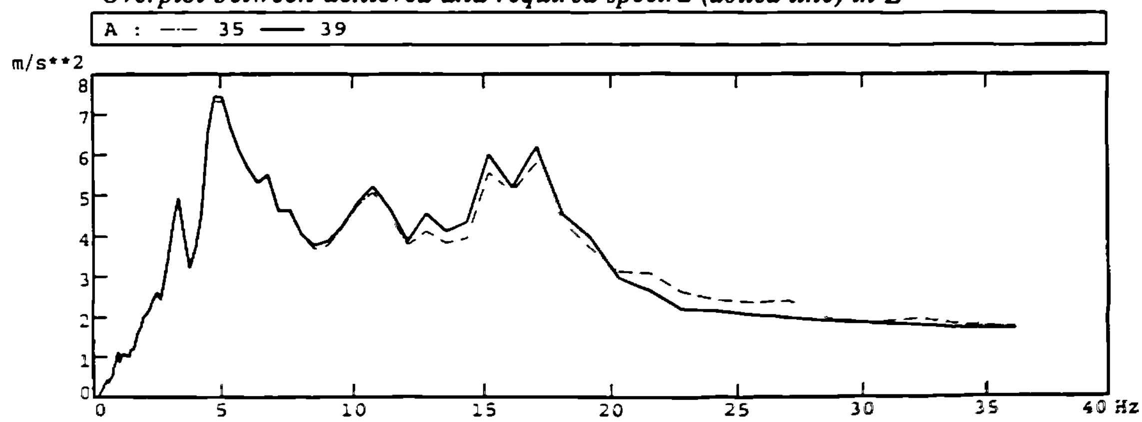
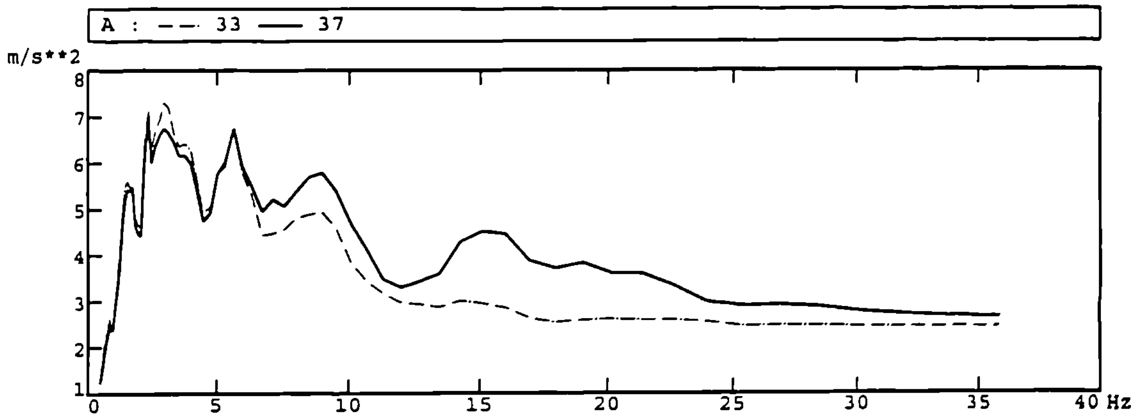


Fig. 5.39 Response spectra achieved on the ISMES table for a 6 DOF displacement match of the Kalamata shake with the 8 tonne flexible specimen

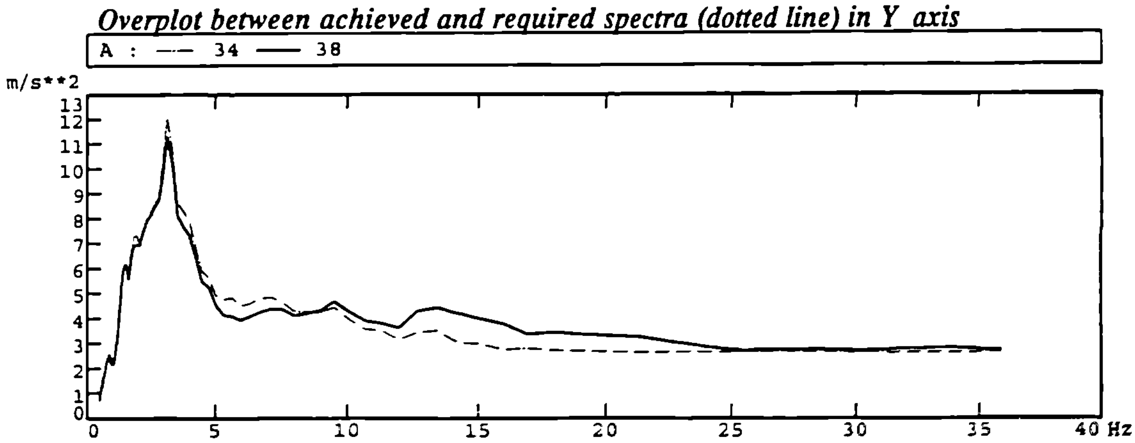
Table conditions : with rigid payload 8t
Controlled DOF : 0
Control mode : acceleration
Drive filter = 0.5 - 35 Hz

Test name : ecoest
Version name : kala8t0
save n.1
Level = 0 dB

Overplot between achieved and required spectra (dotted line) in X



Overplot between achieved and required spectra (dotted line) in Y axis



Overplot between achieved and required spectra (dotted line) in Z axis

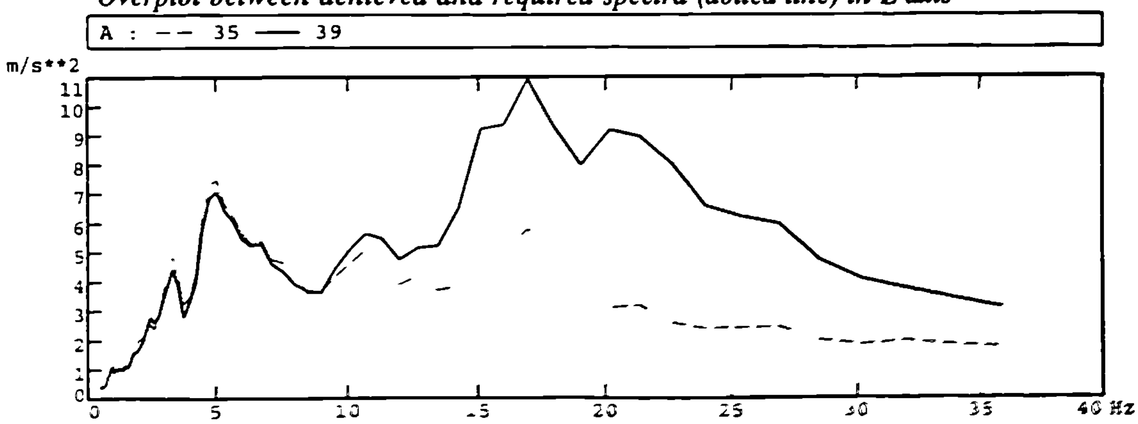
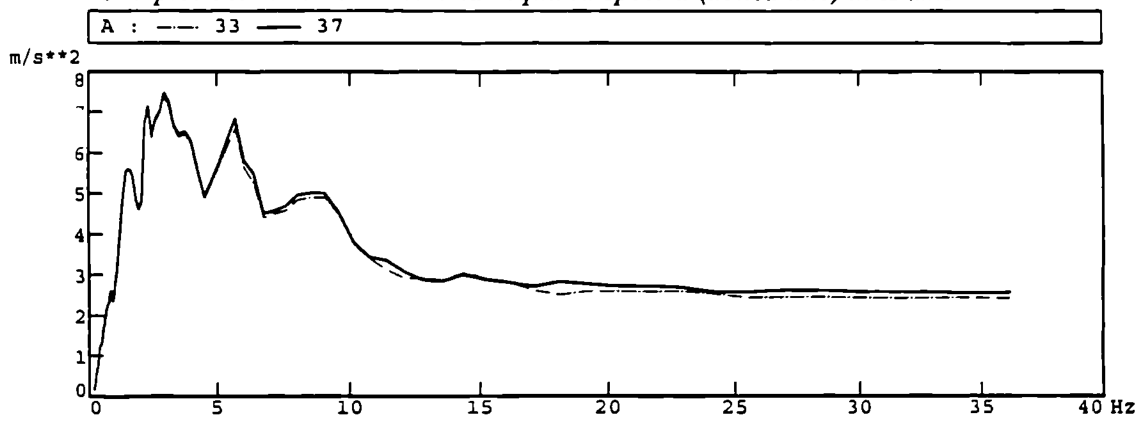


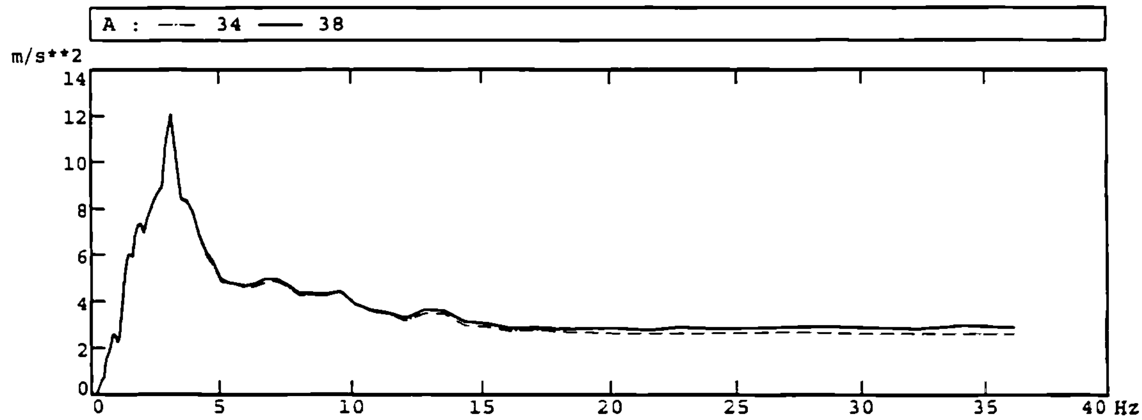
Fig. 5.40 Response spectra achieved on the ISMES table with no software compensation of Kalamata shake with an 8 tonne rigid payload

Table conditions : with flexible payload 8t	Test name : istar
Controlled DOF : 3	Version name : flex3
Control mode : acceleration	save n.6
Drive filter = 0.2 - 35 Hz	Level = 0 dB

Overplot between achieved and required spectra (dotted line) in X axis



Overplot between achieved and required spectra (dotted line) in Y axis



Overplot between achieved and required spectra (dotted line) in Z axis

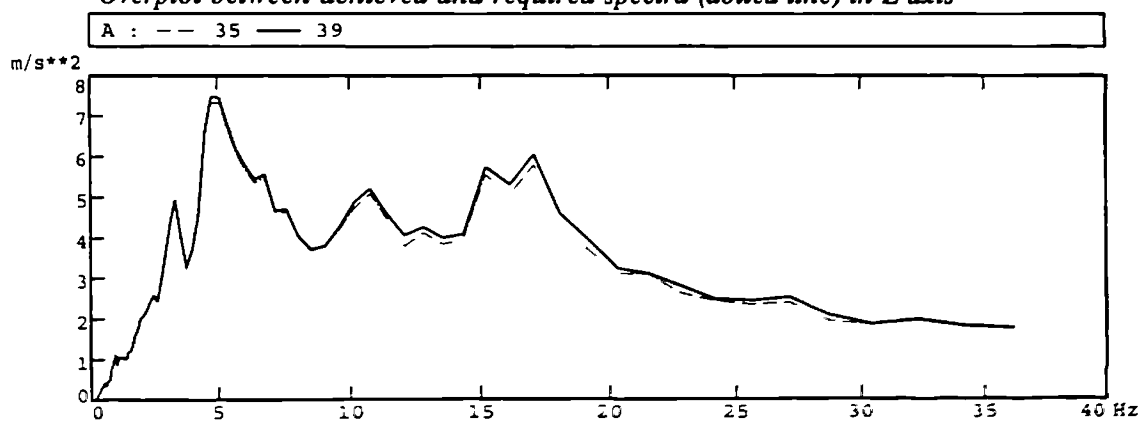
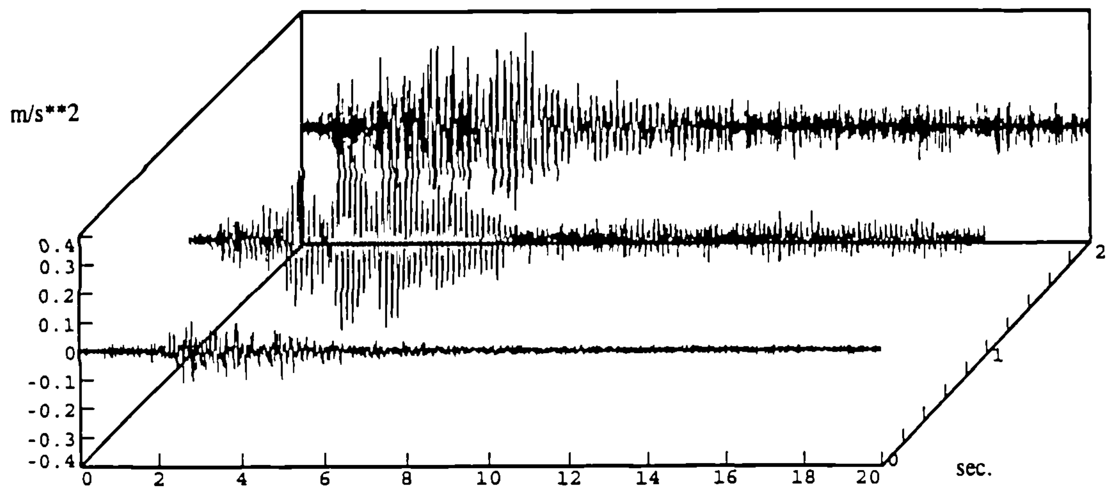


Fig. 5.41 Response spectra achieved on the ISMES table for a 3 DOF acceleration match of the Kalamata shake with the 8 tonne flexible specimen

Table conditions : with flexible payload 8t	Test name : istar
Controlled DOF : 3	Version name : flex3
Control mode : acceleration	save n.6
Drive filter = 0.2 - 35 Hz	Level = 0 dB

Achieved Yaw, Pitch and Roll rms value [m/s**2] : 0.0176, 0.0661, 0.0645

Achieved Yaw, Pitch and Roll time histories



Achieved Yaw, Pitch and Roll FFT

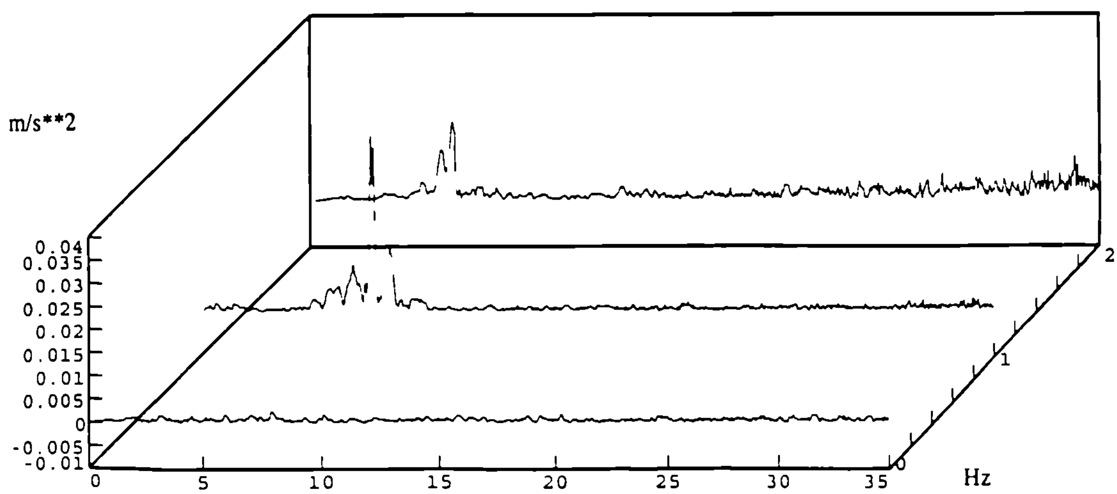
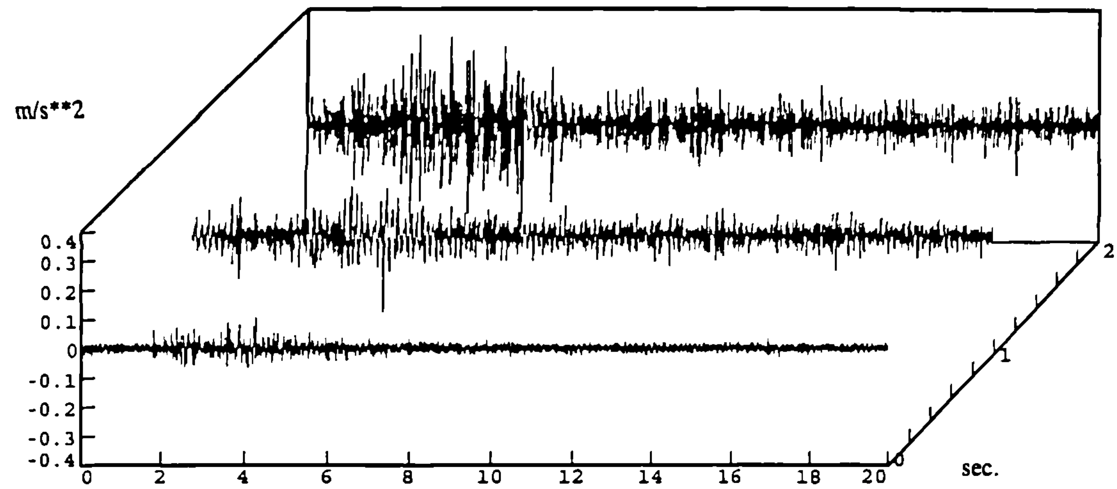


Fig. 5.42 Rotational motions of the ISMES table for a 3 DOF acceleration match of the Kalamata shake with the 8 tonne flexible specimen

Table conditions : with flexible payload 8t	Test name : istar
Controlled DOF : 6	Version name : flex6
Control mode : acceleration	save n.18
Drive filter = 0.2 - 35 Hz	Level = 0 dB

Achieved Yaw, Pitch and Roll rms value [m/s**2] : 0.0155, 0.0373, 0.0608

Achieved Yaw, Pitch and Roll time histories



Achieved Yaw, Pitch and Roll FFT

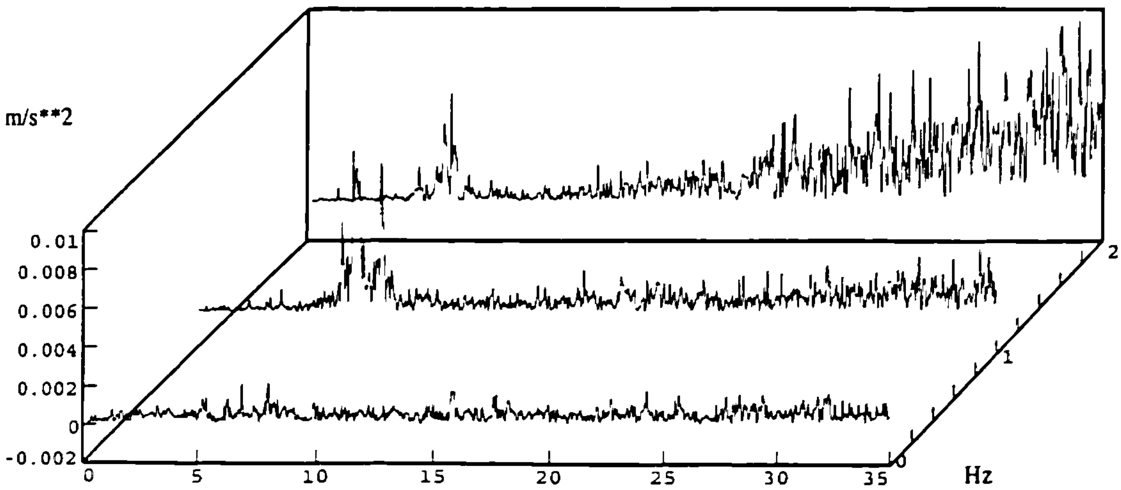
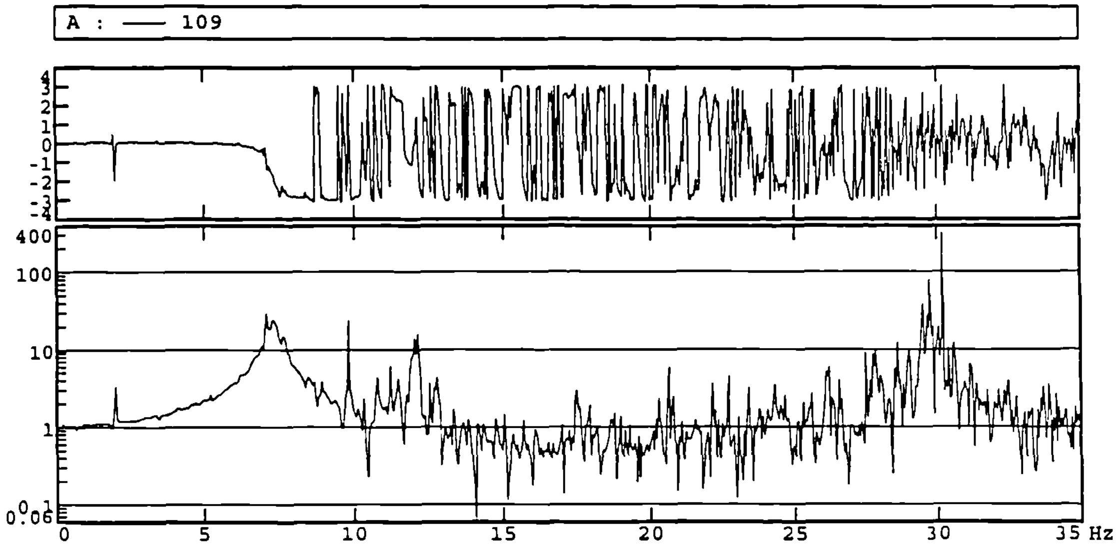


Fig. 5.43 Rotational motions of the ISMES table for a 6 DOF acceleration match of the Kalamata shake with the 8 tonne flexible specimen

Table conditions : with flexible payload 8t
Controlled DOF : 3
Control mode : acceleration
Drive filter = 0.2 - 35 Hz

Test name : istar
Version name : flex3
save n.6
Level = 0 dB

Transfer function between structure and table acceleration in X axis



Transfer function between structure and table acceleration in Y axis

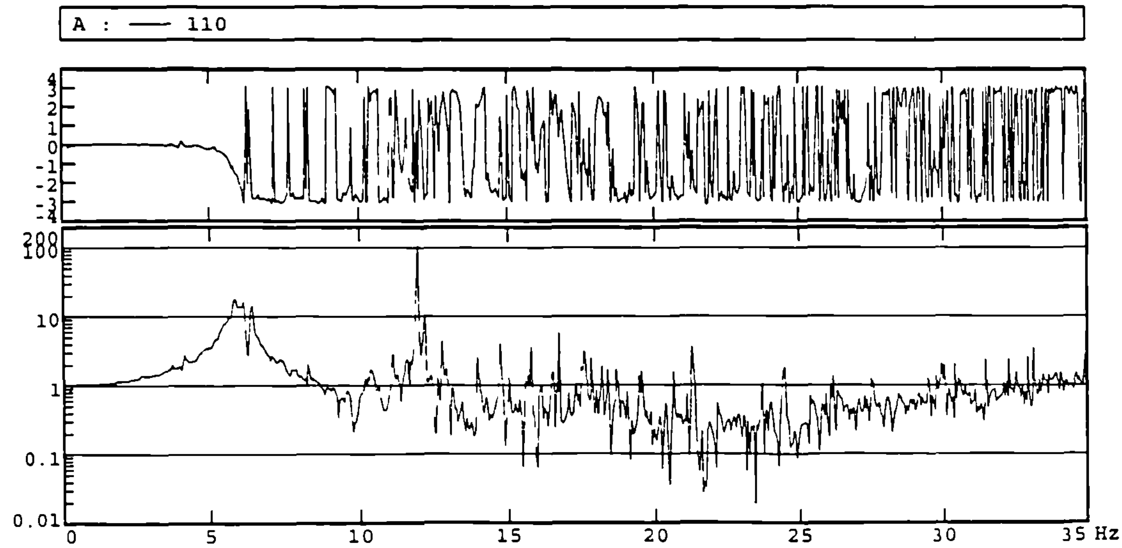
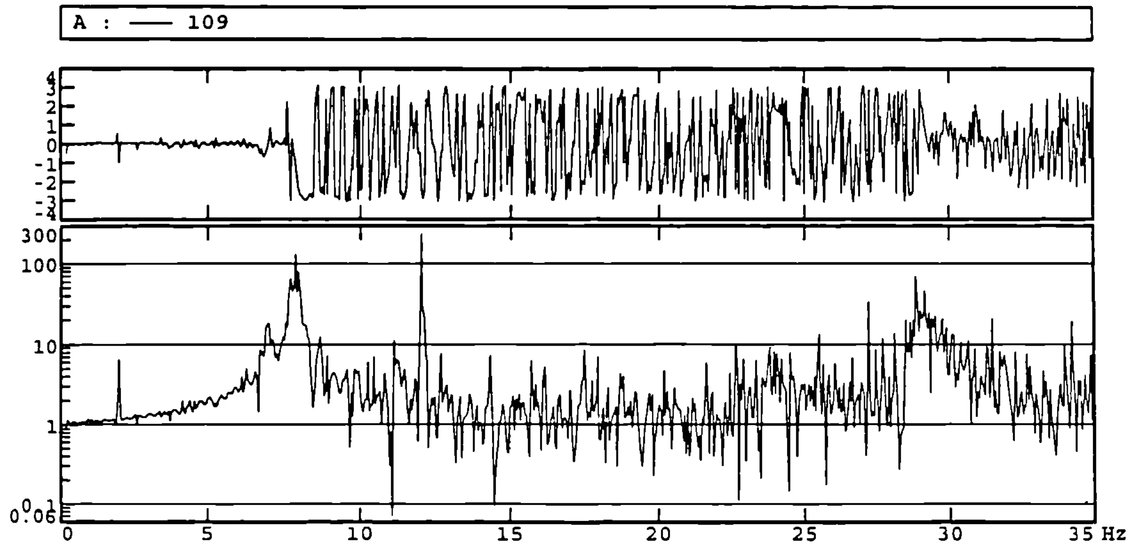


Fig. 5.44 The FRF of the 8 tonne flexible specimen on the ISMES table for a 3 DOF acceleration match of the Kalamata shake

Table conditions : with flexible payload 8t
Controlled DOF : 6
Control mode : acceleration
Drive filter = 0.2 - 35 Hz

Test name : istar
Version name : flex6
save n.18
Level = 0 dB

Transfer function between structure and table acceleration in X axis



Transfer function between structure and table acceleration in Y axis

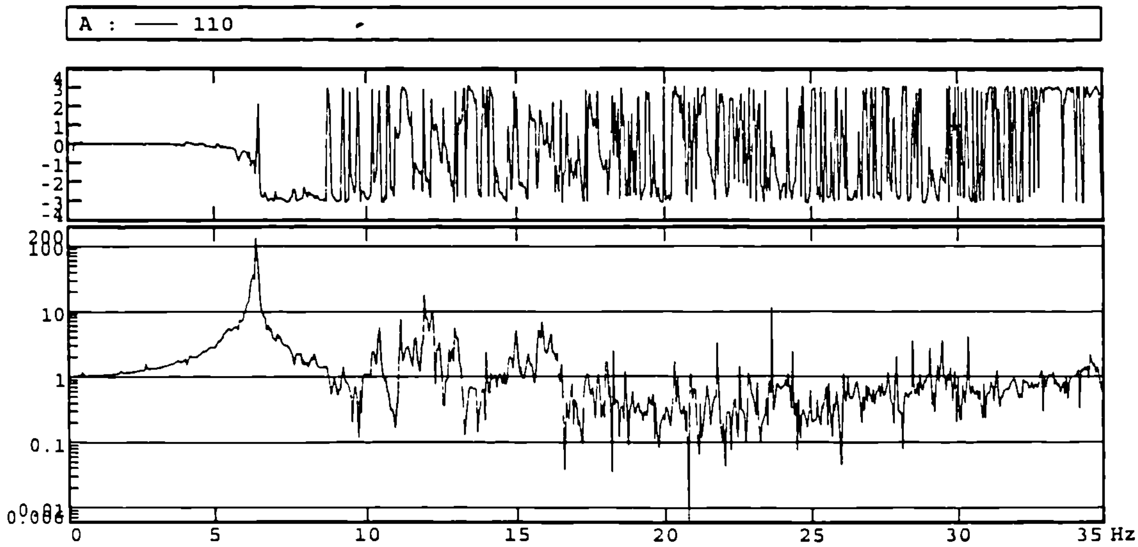


Fig. 5.45 The FRF of the 8 tonne flexible specimen on the ISMES table for a 6 DOF acceleration match of the Kalamata shake

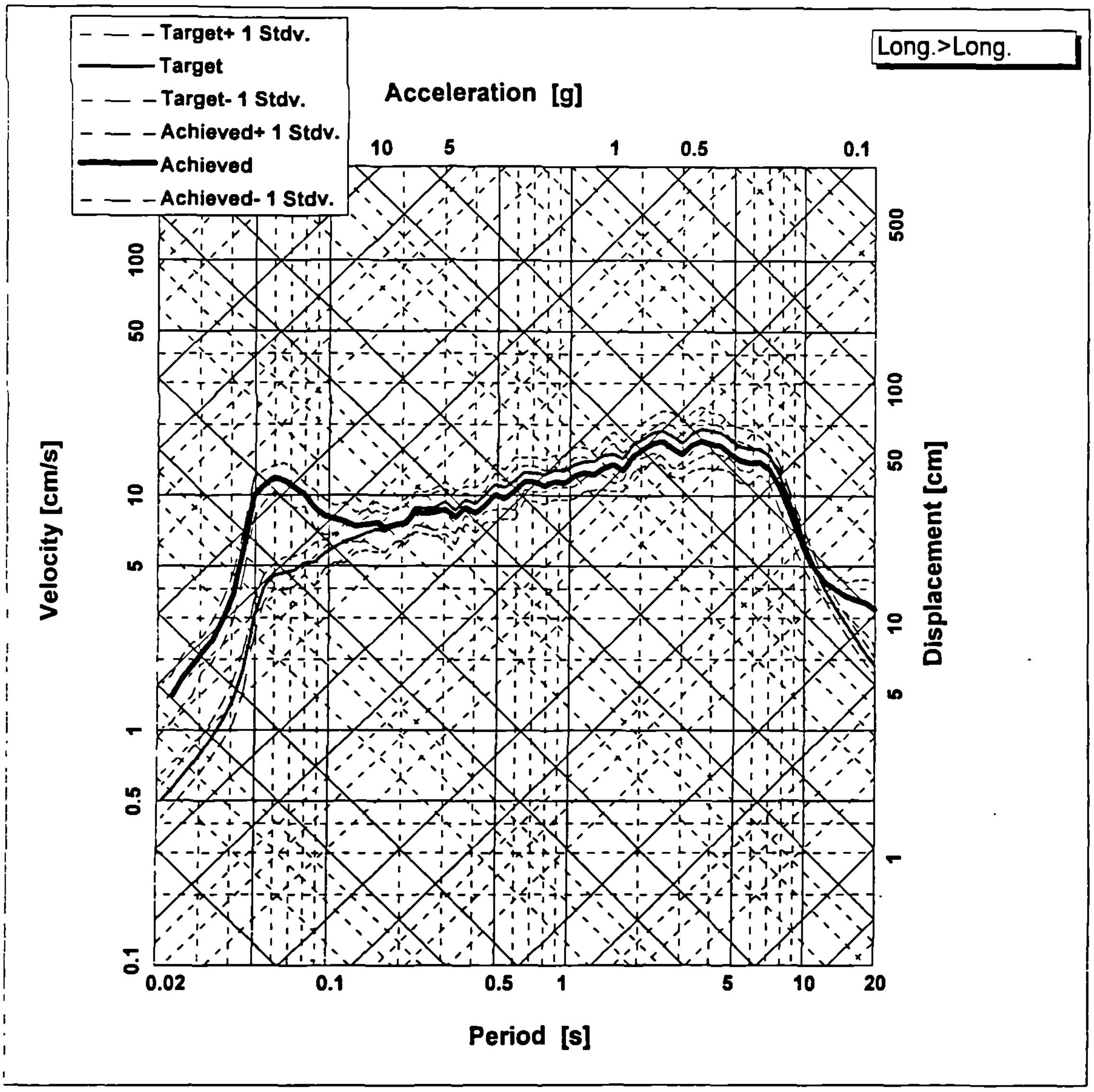


Fig. 5.46 Longitudinal response spectrum achieved on the LNEC bare table for a single-axis match of random white noise

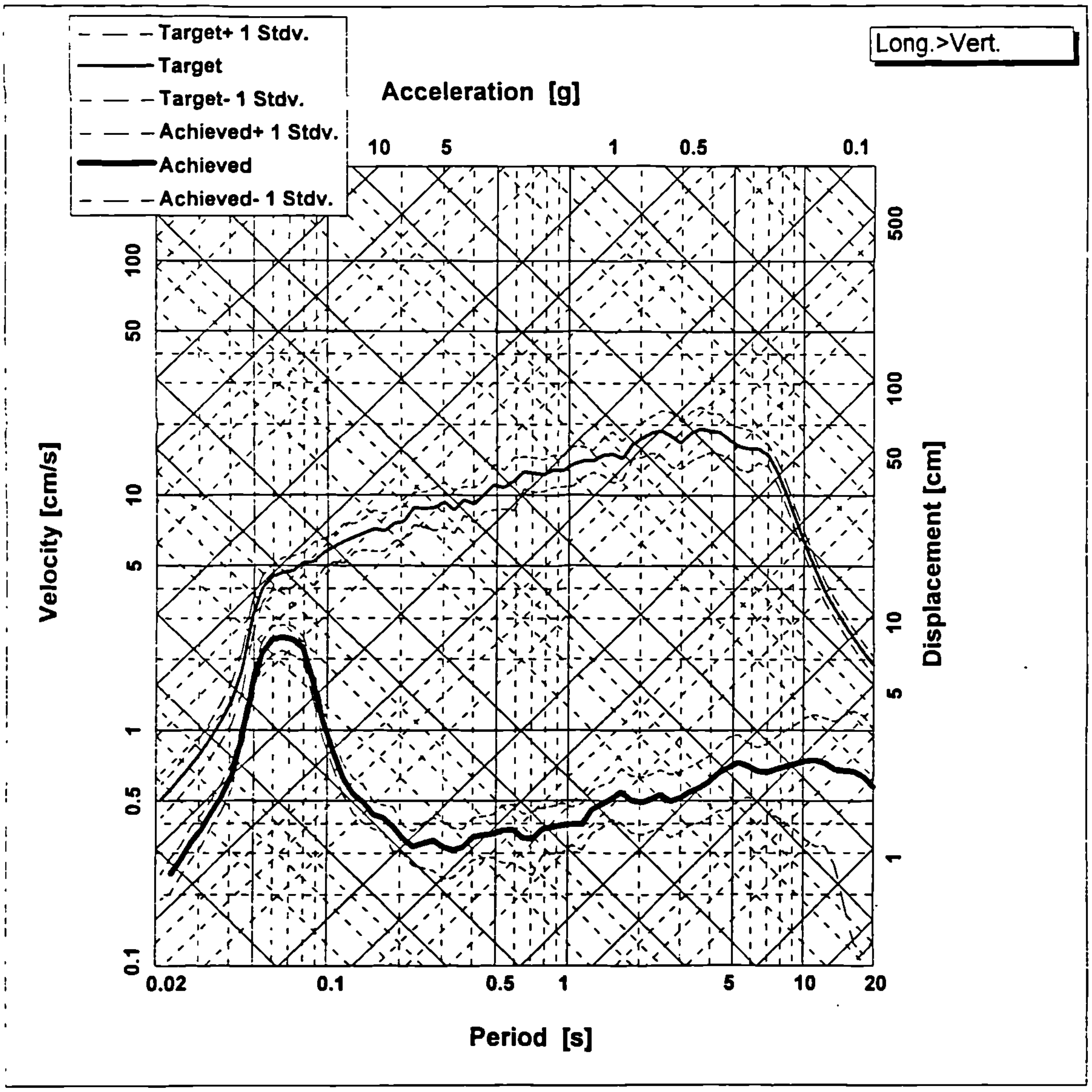


Fig. 5.47 Vertical response spectrum caused by cross-coupling of the LNEC bare table for a single-axis match of random white noise

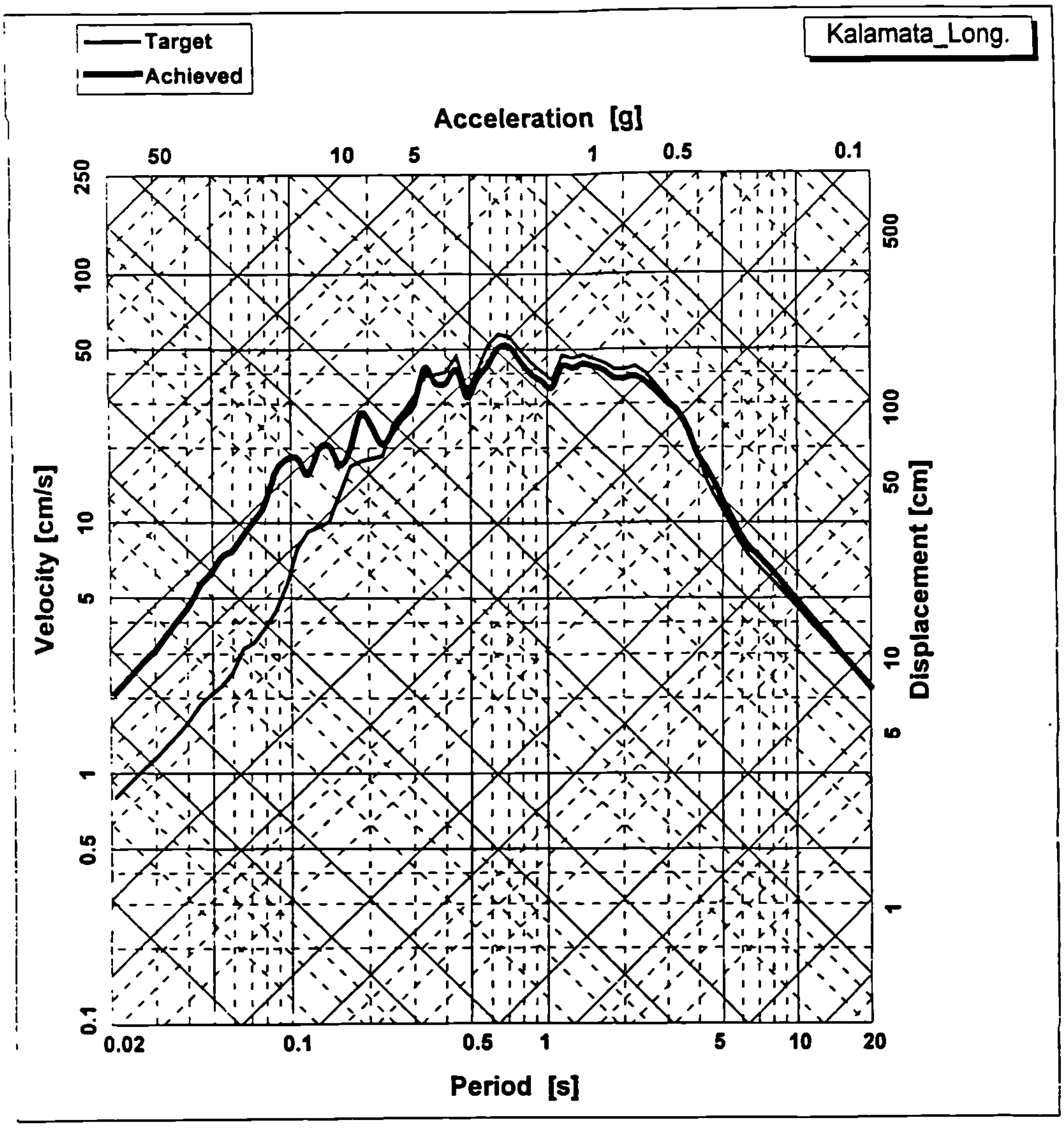


Fig. 5.48 Longitudinal response spectrum achieved on the LNEC bare table for a 3 DOF match of the Kalamata shake

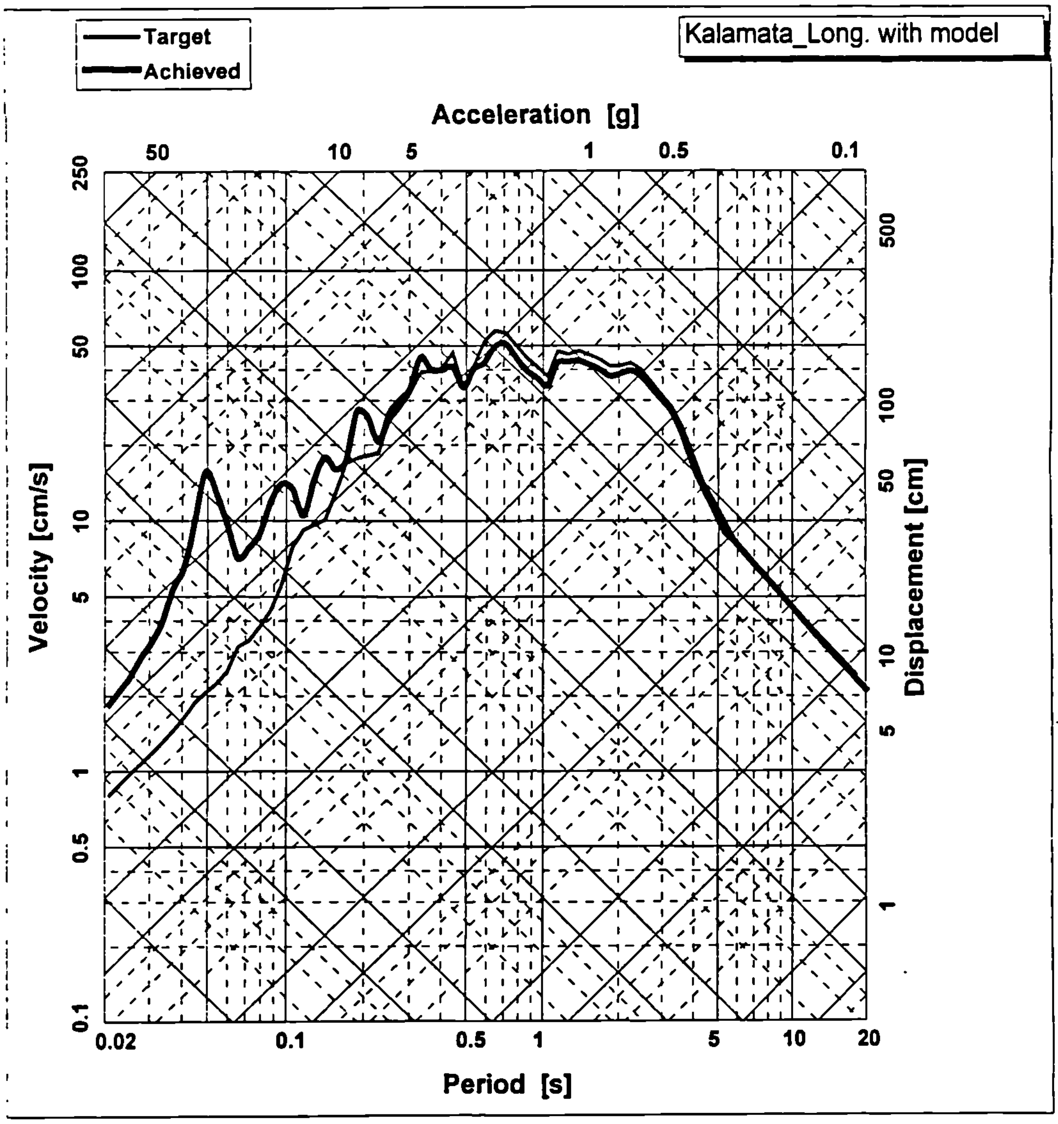


Fig. 5.49 Longitudinal response spectrum achieved on the LNEC table for a 3 DOF match of the Kalamata shake with the 8 tonne flexible specimen

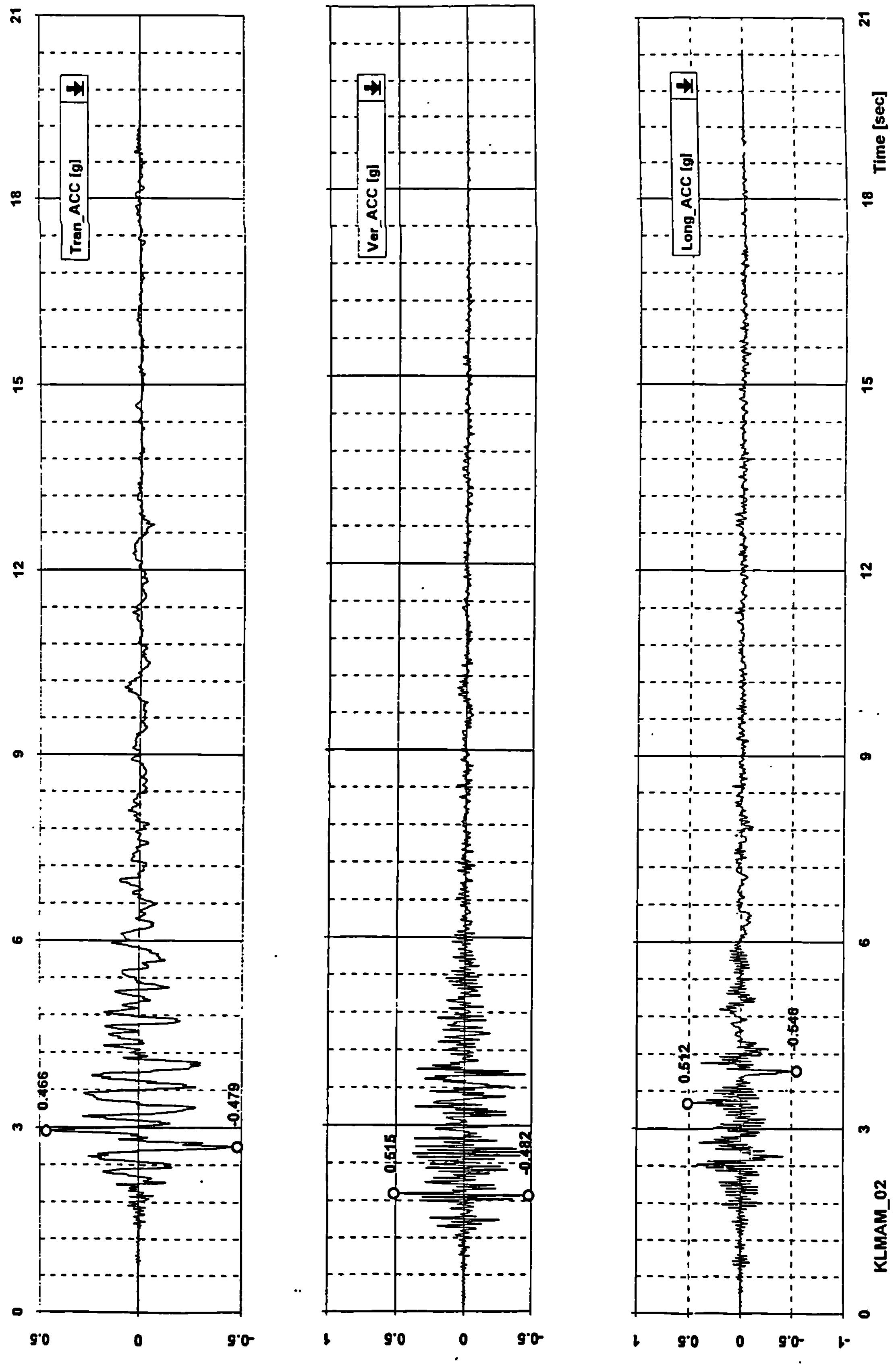


Fig. 5.50 Translational accelerations achieved on the LNEC table for a 3 DOF match of the Kalamata shake with the 8 tonne flexible specimen

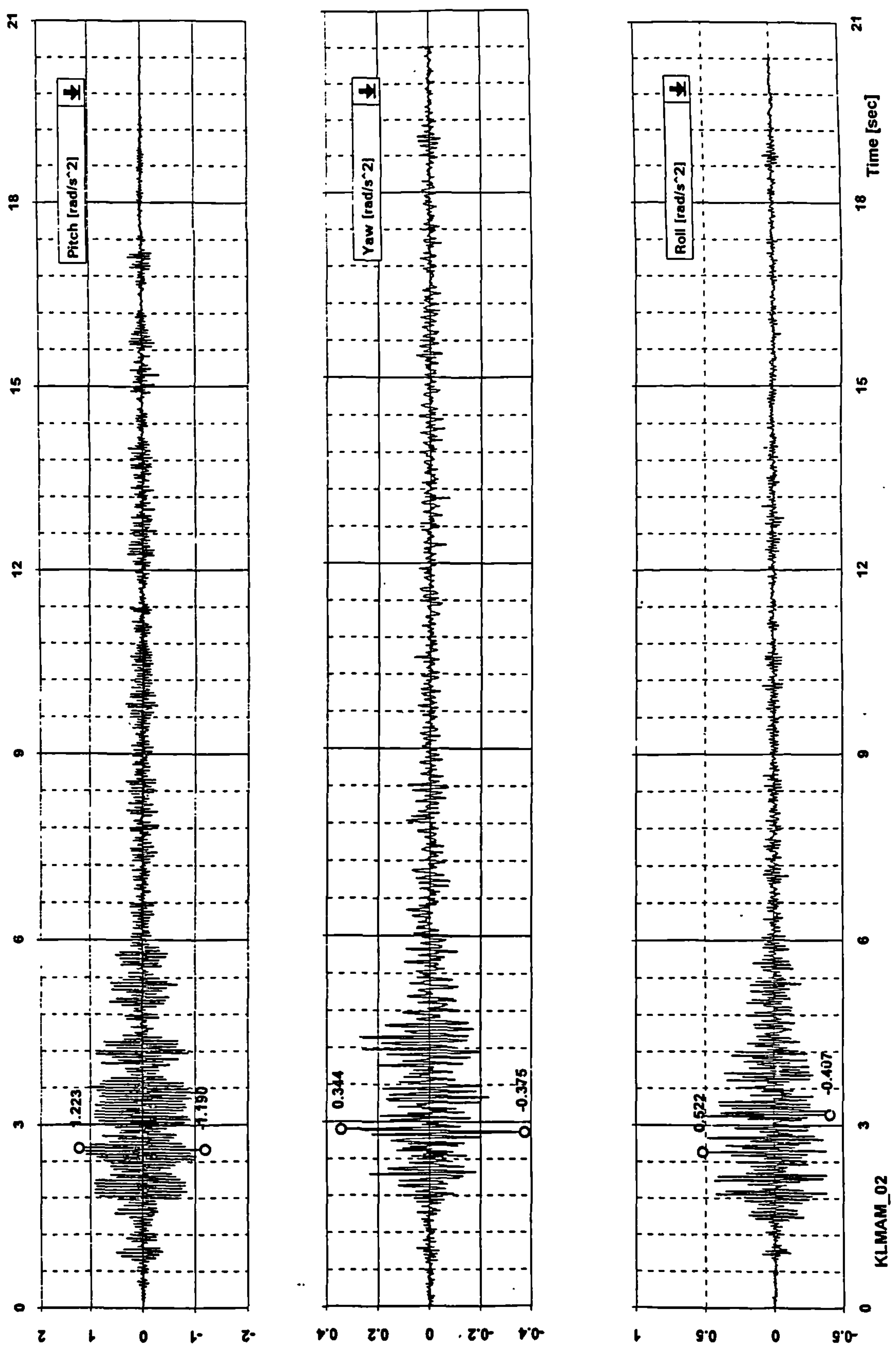


Fig. 5.51 Rotational accelerations recorded on the LNEC table for a 3 DOF match of the Kalamata shake with the 8 tonne flexible specimen

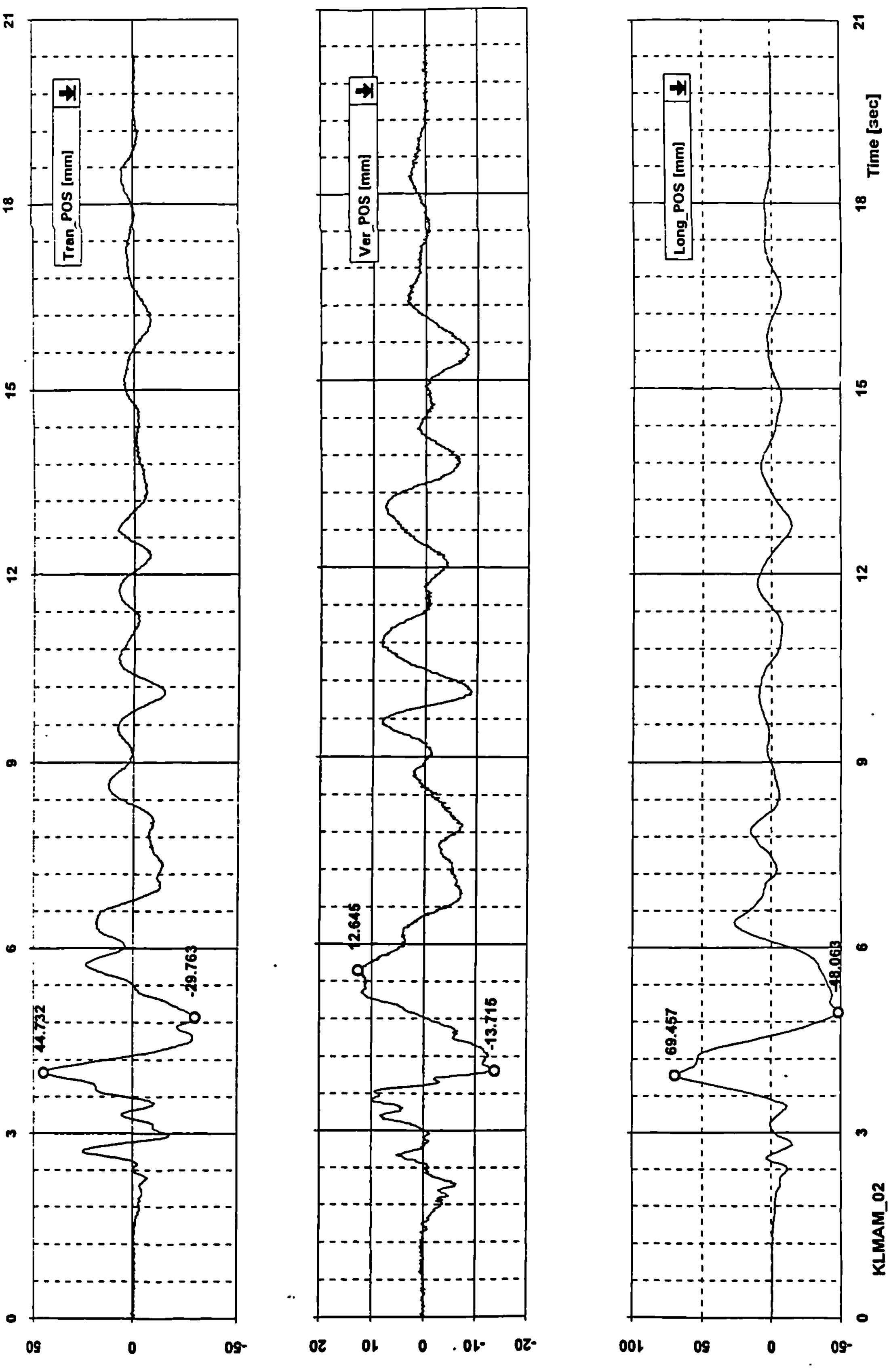


Fig. 5.52 Displacements achieved on the LNEC table for a 3 DOF match of the Kalamata shake with the 8 tonne flexible specimen

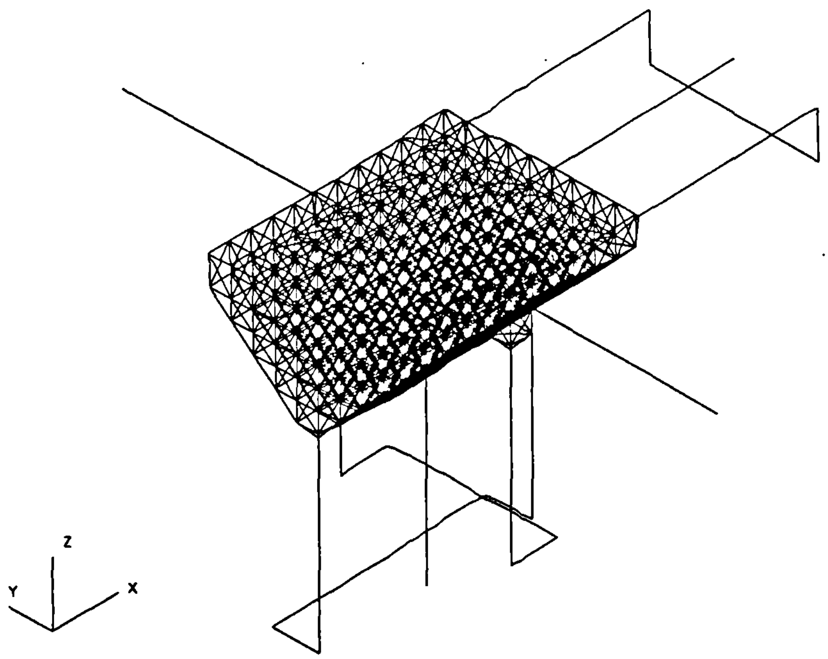


Fig. 5.53 The finite element mesh used in the dynamic analysis of the LNEC shaking table

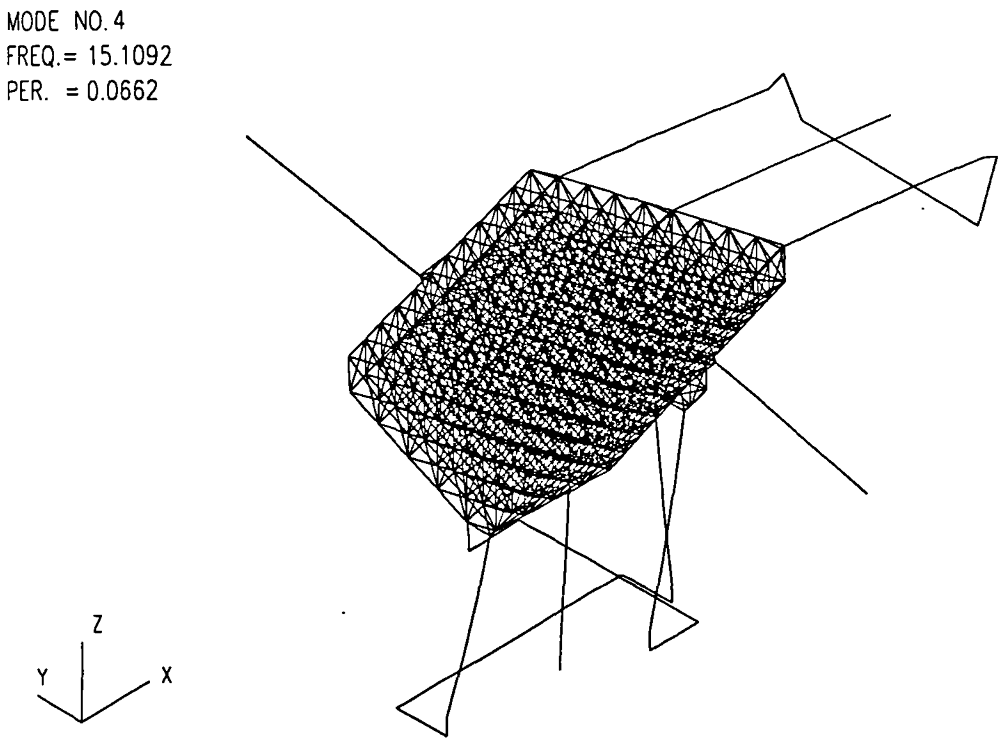


Fig. 5.54 Typical mode shape calculated from the finite element analysis of the LNEC shaking table

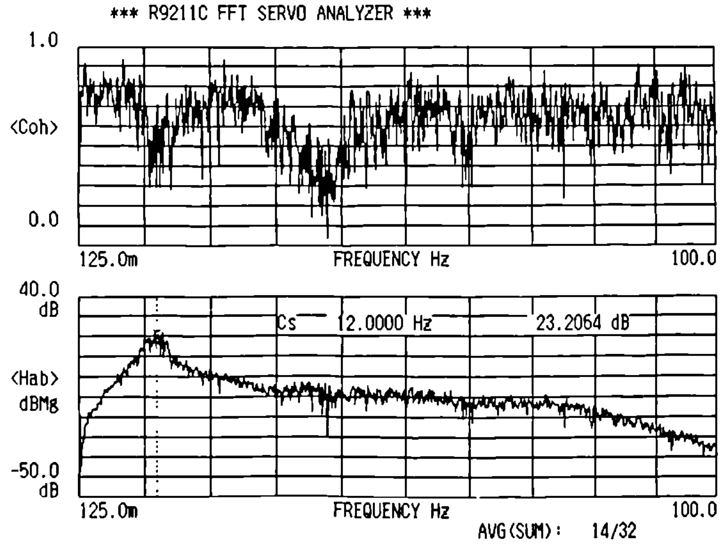


Fig. 5.55a Typical frequency response function of the Bristol table with a 5 tonne flexible specimen attached: without the MCS controller running

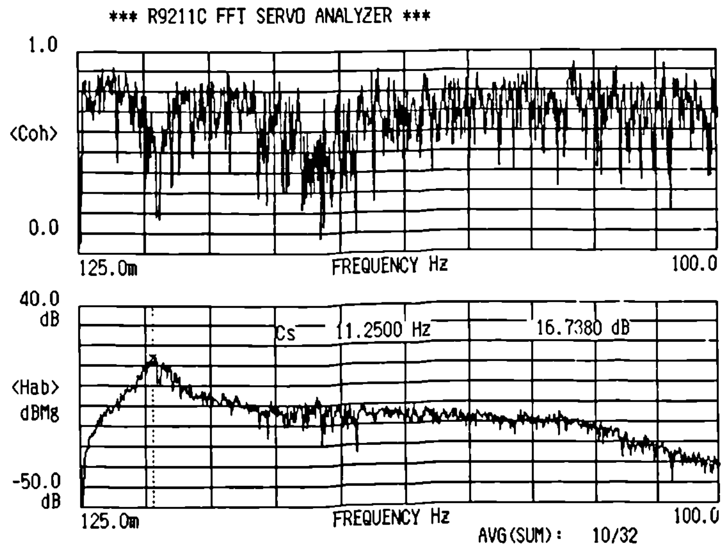


Fig. 5.55b Typical frequency response function of the Bristol table with a 5 tonne flexible specimen attached: with the MCS controller active

(Original in colour)

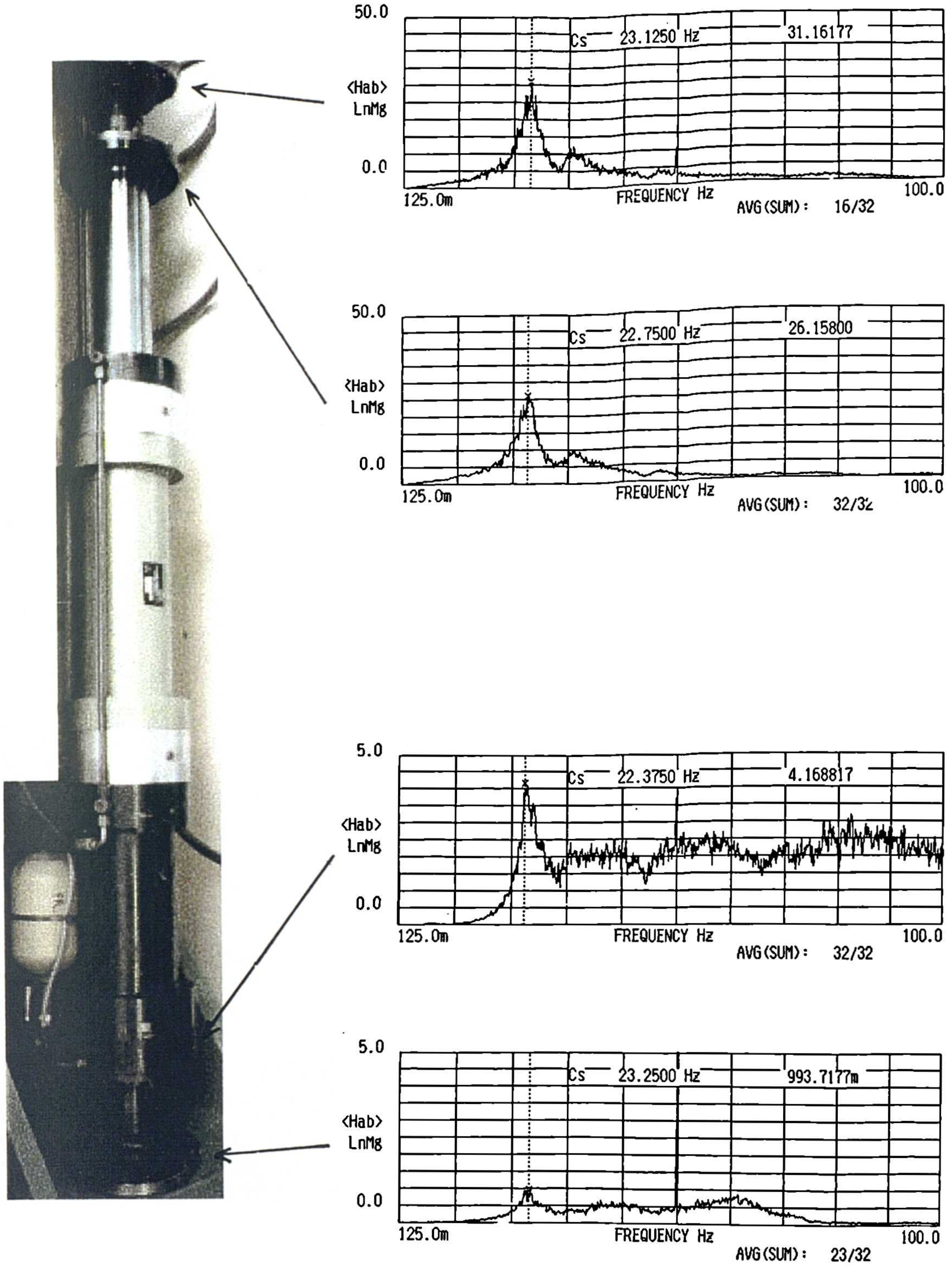


Fig. 5.56 Transfer functions at various points in one of the vertical actuator/bearing arrangements at Bristol

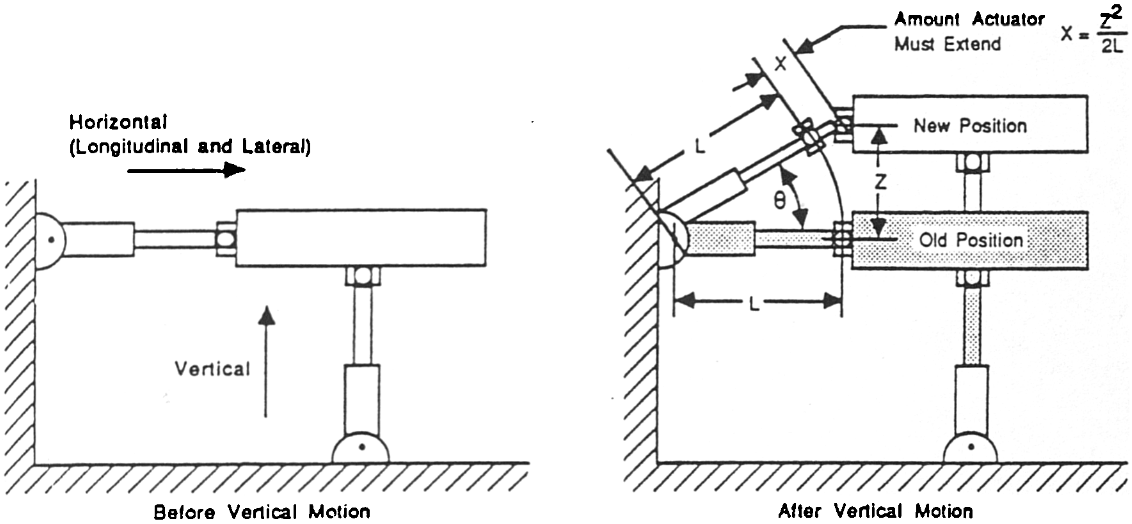


Fig. 5.57 Kinematic model for a two axis table

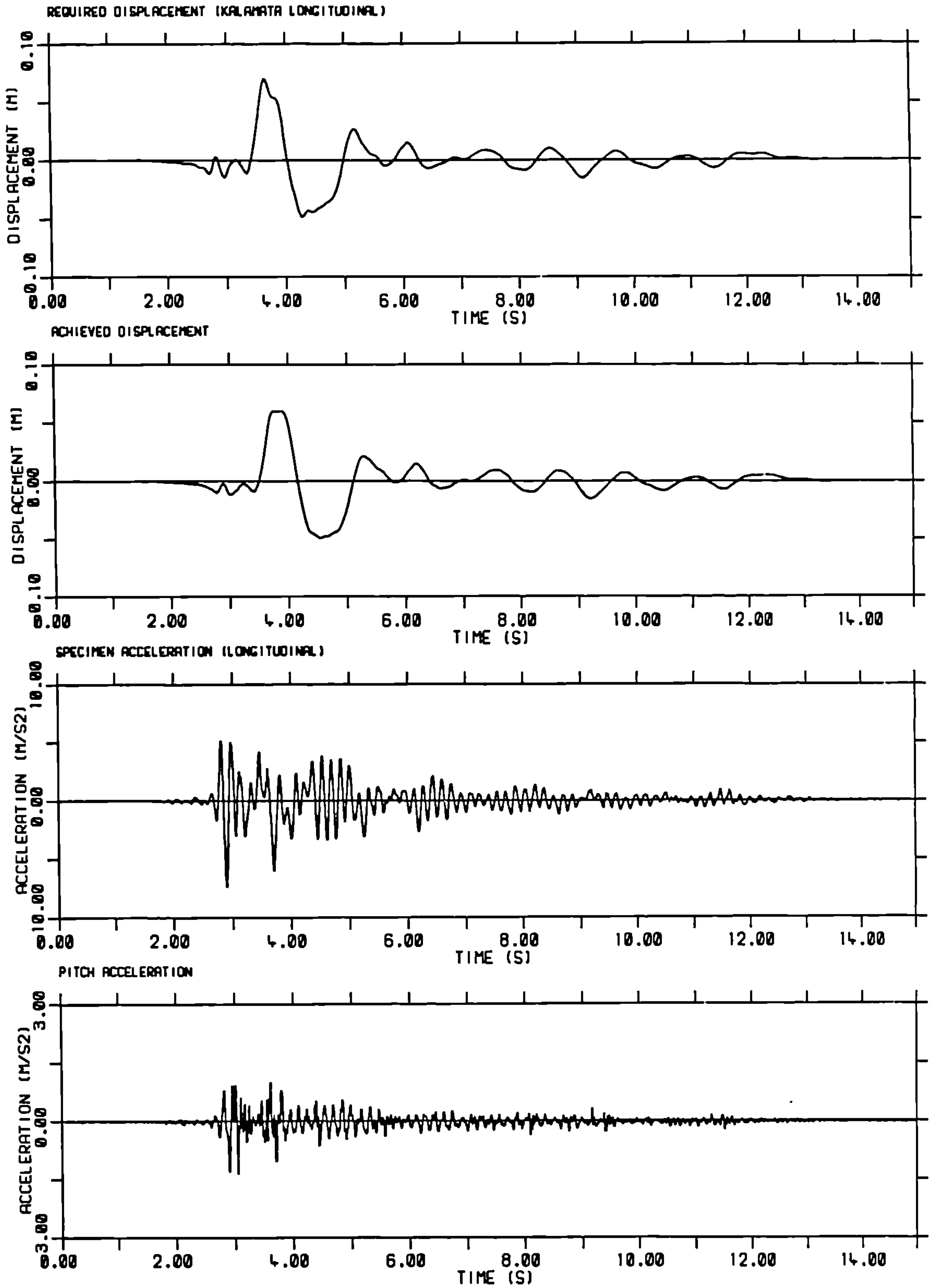


Fig. 5.58 Behaviour of the table and the 5 tonne flexible specimen before matching

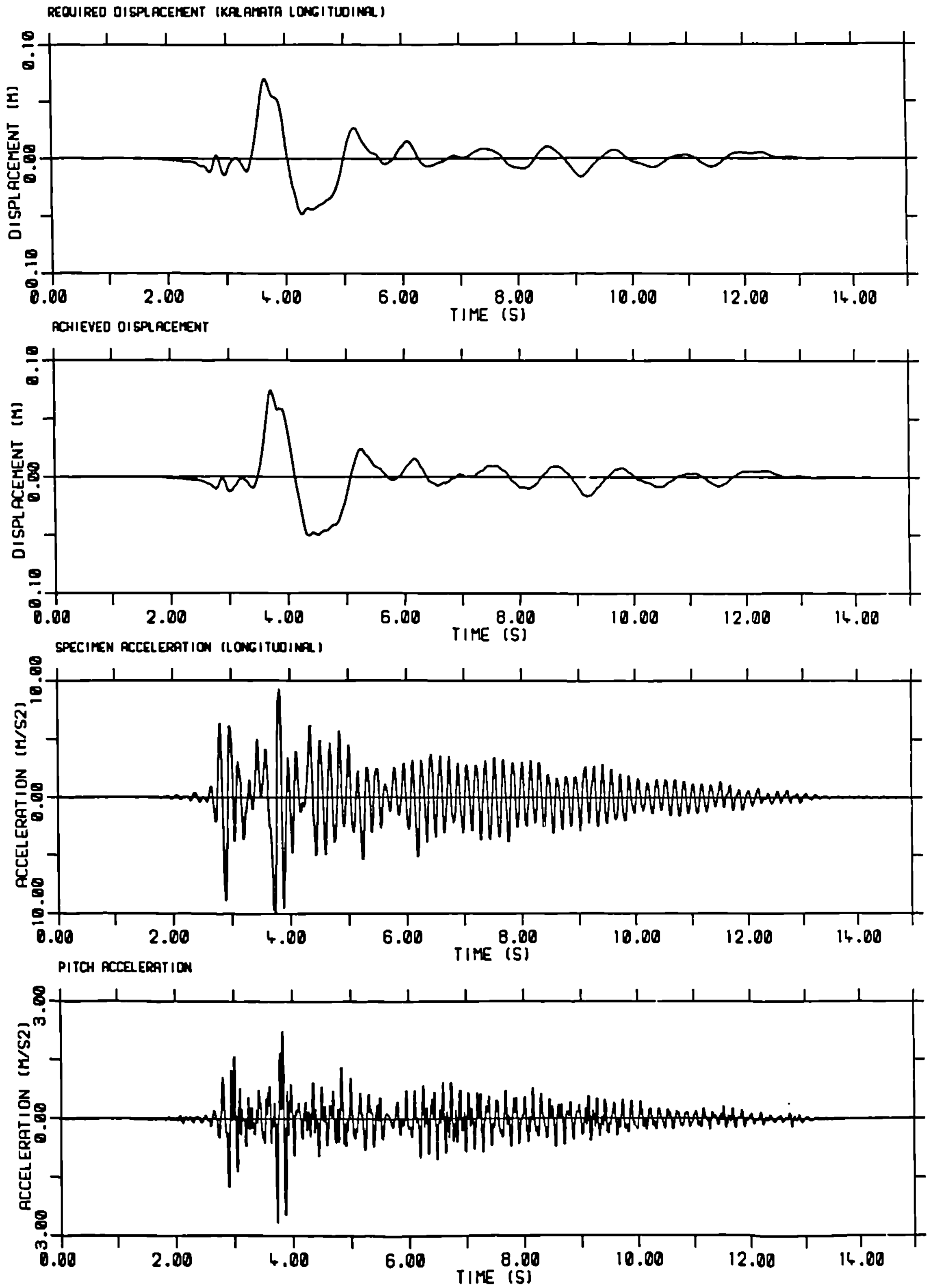


Fig. 5.59 Behaviour of the table and the 5 tonne flexible specimen with MCS running

Chapter 6

Optimisation of Shaking Table Performance

6.1 Introduction

Many of the potential difficulties with the control of shaking tables have been described in Chapter 5. In this chapter several new testing methods that were developed during this research programme are described, and examples are given of their effectiveness in improving the accuracy of the platform motion during shaking table tests. A few techniques that may significantly improve the use of shaking tables in the future are also briefly outlined, together with some of the initial results of the development tests.

6.2 Shaking table hardware

6.2.1 Mechanical characteristics

The main mechanical problems that can cause resonances in any shaking table system were discussed in §4.3.2. Of all these problems the two that caused most difficulty at Bristol were the backlash in the spherical bearings (§4.3.2.4) and the oil column resonance (§4.3.2.6). The problems of backlash in the bearings are minimised by employing a regular maintenance schedule and by performing a careful inspection of the table before each test is performed. After the initial tests at Bristol and Athens, which highlighted the advantages of pressure transducers across the actuators to help tuning and minimise actuator fighting, differential pressure transducers were subsequently installed across all actuator manifolds at Bristol. Apart from greatly simplifying the tuning of the table, these transducers, being very sensitive to very small displacements and very high frequency loading, are also used to monitor any high frequency shocks occurring through any of the actuators which might indicate that the bearings at one or both ends of the actuator need additional maintenance. The accelerometers at the ends of the actuators may, in a similar way, be used to monitor any backlash in the bearings, but they are slightly less sensitive to the forces caused by any bearing backlash.

The oil column resonance is a more significant problem at Bristol. Although the table control software is able to compensate for these resonances to a certain extent, because the resonances are only very lightly damped, there is the possibility that, in certain circumstances, the iterative matching software can become unstable as it tries to control these resonances (§5.5.1.2). The oil column resonance may also be having a detrimental effect on the stability of the MCS algorithm (§6.7) when it is attempting to control the table in real-time. Work is currently in progress to find the most appropriate technique to use in Bristol reduce these resonances. Several options are currently being considered. Kusner et al. (1992) suggest that, based on data from field tests, incorporating some form of servo-valve linearization can reduce distortions caused by the oil column resonance by up to 50%. MTS Inc. control this resonance by incorporating a 'Three Variable Control' system in the hardware of the shaking tables they design (Clark, 1992). A final possibility is the incorporation of damping elements across the actuator manifold between the oil on the two sides of each actuator piston, although this option may reduce the overall dynamic capacity of the actuators. It is hoped that, using one or more of these techniques, the oil column resonance can be significantly reduced, which should solve the control instabilities that can occur with the various software control techniques in use at Bristol.

If the performance of a shaking table is being significantly affected by any of the other problems mentioned in §4.3.2, then more significant modifications to the table will be required. For example, at the start of the test programme in LNEC it was noticed that there were significant local resonances in the torque tube system. Directly as a result of this research, two different remedial actions were taken at this site. In the first instance additional dampers were attached to the linkages connecting the torque tubes to the platform to damp out the lateral bending modes that were occurring (§4.3.2.7). In the second instance, a dense expanding foam was injected into many of the linkages forming the restraining system, which damped out the local plate resonances. While neither of these actions eliminated the resonances, they did significantly reduce the scale of the problems; and the software control system was subsequently able to compensate for the resonances much more effectively.

6.2.2 Tuning of the hardware control system

The tests at all four laboratories highlighted the importance of tuning the shaking table control hardware as accurately as possible. Although this in itself is not sufficient for good overall system performance, if the table transfer functions are essentially flat this will minimise the work that the software control has to do, and reduce the likelihood of instabilities occurring during the iterative matching process. However, the shaking table user should always be aware that, regardless of how good the system tuning is, additional software control will always be needed to cope with any non-linear specimen-table interaction.

The methods of tuning a shaking table will always vary between laboratories because of the different hardware at each site, and because each hardware system will have different forms of feedback that can be adjusted to compensate for resonances in the shaking table. However, the principles for tuning a shaking table will be the same at all laboratories. The process should start with the measurement of the transfer function for the table axis / actuator being tuned. If possible, this should be performed using a broadband random noise signal and a spectrum analyser that can give an instantaneous display of the transfer function or power spectrum of the platform motion. By using an instantaneous display, it is then possible to see the effect of any adjustment of the hardware immediately. This greatly simplifies the whole tuning process, as in some cases there may be up to eight gain terms to adjust for each table axis. The various gains should then be adjusted to give the flattest platform transfer functions. The effectiveness of this process will generally depend on the skill of the shaking table operator, although it should be noted that each of the main feedback gains affect different frequency ranges in the overall transfer function in different ways. Use of the table below to compare the frequency of the worst error with the gain to be adjusted, should allow an inexperienced user to tune a shaking table more quickly than simply by resorting to trial and error.

It can be seen from table 6.1 that control hardware that has many feedback terms (acceleration, velocity, displacement, pressure etc.) can allow much better adjustment of the table performance than hardware with just displacement and acceleration feedback. This is also seen by comparing the results of the tuning process at Bristol (very simple feedback) and ISMES (all the feedback terms described above). However, it should be noted that the tuning of up to 52 individual gains – many of which cannot be adjusted in isolation – is a very time consuming process (possibly taking several hours).

Table 6.1 Effect of adjusting various hardware feedback gains

Feedback adjustments	**Effect on table performance in frequency domain**
For each actuator	
Servo valve gain	Adjusts all frequency components up or down. This is normally adjusted by the table manufacturer for optimum performance of the servo valve and should not be adjusted by the shaking table user.
For each actuator/axis	
Master gain	Adjusts all frequency components up or down.
Displacement gain	Adjusts the low frequency components up or down (0 Hz to ≈ 5 Hz).
Velocity gain	Adjusts the medium frequency components up or down (≈ 4 Hz to ≈ 12 Hz).
Acceleration gain	Adjusts the high frequency components up or down (≈ 10 Hz to ≈ 100 Hz).
Delta Pressure / Force gain	Adjusts the very high frequency components up or down (≈ 40 Hz to ≈ 100 Hz) and at the same time lowers and raises the lower frequency components (0 Hz to ≈ 40 Hz) respectively. i.e. provides rotation of the frequency components about a frequency point of ≈ 40 Hz.
Velocity Lead gain	Adjusts the medium frequency components up or down (≈ 4 Hz to ≈ 12 Hz). This should be used if further adjustment of these frequencies is required beyond the point when increasing the velocity gain further causes table instability.
Acceleration Lead gain	Adjusts the high frequency components up or down (≈ 10 Hz to ≈ 100 Hz). This should be used if further adjustment of these frequencies is required beyond the point when increasing the acceleration gain further causes table instability.
Servovalve shaping gain	Adjusts the high frequency components up or down (≈ 10 Hz to ≈ 100 Hz).
For whole table	The three adjustments below provide additional compensation for table-specimen interaction by feeding various amounts of the desired translational motions directly into the rotational axes.
Roll compensation gain	Feeds a proportion of the Y axis signal into the Roll axis. This compensates for a specimen with a high centre of gravity being shaking in the Y axis.
Pitch compensation gain	Feeds a proportion of the X axis signal into the Pitch axis. This compensates for a specimen with a high centre of gravity being shaking in the X axis.
Yaw compensation gains	Feeds proportions of the X and Y axis signals into the Yaw axis. This compensates for a specimen which is not placed at the centre of the table being shaking in the X and / or Y axes.

6.3 Shaking table software

6.3.1 Use and development of existing iterative matching software

Several different methods of iteratively matching time histories were used during this research programme, and the advantages and disadvantages of each method were fully explored. In most cases the simpler matching techniques performed well (i.e. at Bristol, §5.5.1.2 and at Athens, §5.5.2.2), but the more sophisticated techniques developed as a result of these tests generally performed slightly better (i.e. at ISMES, §5.5.3.2 and at LNEC, §5.5.4.2) as can be seen from the results presented in Chapter 5. Based on the experience of these tests, the following section outlines the different techniques used at the four sites, including the new techniques developed during this research programme, along with the circumstances under which they were found to perform most effectively.

Drive signal precompensation – Although this method of drive signal correction does not involve any iteration, it has been shown to improve significantly the accuracy of platform motions resulting from the first attempt at any matching. This precompensation will also reduce the number of subsequent iterations that are needed to produce an accurate platform response. Before any testing starts, the table (with any specimen attached) is driven in all axes with a random white noise signal and the actual motions of all the table axes are recorded. By comparing the drive and recorded signals with each other the transfer function of the entire shaking table system can be calculated. The desired platform motions are then multiplied (in the frequency domain) by the inverse system transfer function to produce a much better first attempt at a drive signal that will produce the actual platform motion desired (Flesch, 1986). Figure 6.1 shows a first attempt a reproducing the lateral Kalamata motion on the ISMES table without any drive signal precompensation, while figure 6.2 shows the improvement that occurs if the drive signal is precompensated to take the system transfer functions into account. While there are clear advantages in performing this precompensation, one of the difficulties is that the loading on the platform should be exactly the same as for the actual test. Then, if the amplitude of the white noise is too high when the transfer functions are being measured, the specimen on the platform may be damaged. However, if the amplitude of the white noise is too low, then non-linearities in the amplitude scaling of the drive signals may mean that the transfer functions generated from the low amplitude white noise excitation are not representative of the actual system performance with large amplitude earthquake signals. Following the tests at the four laboratories it is recommended that, if possible, the first iteration of any matching should include this precompensation stage. Even if only low level signals can be used to calculate the transfer function, the modified drive signals are still likely to be better than the unmodified signals as a first stage in the matching process.

Transfer function updating – In this method the software calculates a new transfer function matrix from the drive signals and the actual platform motions after each iteration of the matching. The new set of drive signals are then obtained from the desired signals and the new transfer function matrix. This method is very simple, being in essence a repetition of the drive signal precompensation technique. However, because the drive signal does not necessarily have a uniform frequency content across the operating frequency bandwidth of the table, the user must be very careful when performing this type of iterative matching. If some parts of the frequency bandwidth of the drive signal have little energy, the new transfer function will be less accurate over this frequency range and the correction process is likely to become unstable. This iterative matching technique is therefore only appropriate when the desired platform motion has a reasonably broad frequency content.

Iterative spectral matching – In this method the software calculates the spectra of the actual platform motions, compares them with the spectra of the desired platform motions, and modifies the amplitude of the frequency components of the drive signals for the next iteration based on the ratio between the achieved and desired spectra. This technique can also be combined with the process of transfer function updating to improve the stability of the iterative matching. It should be noted that this method can modify the drive signals so that they may go beyond the capacity of the table, and that this system of iterative matching was found to be inappropriate if a specific drive signal was to be recreated on the platform. This method is best used when a test requires that a spectrum is matched, rather than an actual earthquake motion. This type of iterative matching should be used to create the platform motions used in seismic qualification testing (§2.2.5).

Iterative linear time history matching – In this method the software calculates the ratio, in the frequency domain, of the actual platform motions and the desired platform motions (figure 4.27). The amplitudes of the frequency components of the drive signals are then modified in proportion to this ratio (figure 4.28a) for the next iteration. The time histories may also be segmented into windowed overlapping blocks, the inverse transfer function for each block computed, and an updated drive signal for each block created. This approach leads to a time dependent transfer function (§4.4.2.2) that can deal with repeatable non-linearities in the shaking table system itself (such as a drop in oil pressure during the test) but at the expense of the frequency resolution of the transfer function. The time history tests at Bristol (§5.5.1.2) used segmented time histories, while those performed in Athens (§5.5.2.2) worked on the entire time history at once. The loss of frequency resolution can be seen by comparing figure 5.12 (Bristol) with figure 5.22 (Athens). The frequency resolution at Bristol was only 0.391 Hz whereas at Athens it was 0.024 Hz. These values are also the minimum frequencies that will be matched by the software, and a drop in amplitude of the transfer function between the actual and the desired platform motion below these frequencies can be seen in the two figures. In particular, figure 5.12 shows a significant error in the matching below 0.4 Hz on the Bristol table. The frequency resolution (Δf) can be calculated from:

$$\Delta f = \frac{1}{\Delta t . N} \tag{6.1}$$

where N is the number of samples in the segment of the time history and Δt is the sampling time interval. The segmented approach to matching time histories iteratively can also cause instabilities in parts of the signal where there is little energy; for example, in figure 6.3 the platform accelerations recorded show significant noise, cause by such numerical instability in the matching algorithm, between 8 and 10 seconds into the earthquake. The use of the non-linear iterative matching method described below can reduce this instability in most cases. However, while these problems occasionally cause difficulties, the benefits of coping with repeatable non-linearities in the shaking table system often outweigh the problems caused by analysis of a segmented time history. The shaking table user should be aware of the many analytical options that are available when performing this type of iterative matching, and should use appropriate coefficients in the software algorithms for the type of test being performed.

Iterative non-linear time history matching – This method is identical to the previous one, apart from the way the frequency components of the drive signals are modified after each iteration of the matching process. In this case the frequency components of the drive signals are modified by the addition of an arbitrary proportion (normally about 70%) of the FFT of the compensated error between the actual and desired platform motions. The error signals can be compensated in two ways. In the first case the initial transfer function matrix (created at the drive signal precompensation stage) is used to correct the error signal and compensate for the system transfer function. In the second case the error signal is corrected using a revised transfer function matrix calculated from the last iteration (figure 4.28b). This second method is basically a combination of transfer function updating and non-linear iterative matching. As the drive signals are not modified quite so rapidly using either of these techniques, the process is more stable when iteratively matching a desired signal that has a lower energy content than are the linear matching methods. Non-linear iterative matching techniques should therefore be used when the linear matching algorithm shows signs of numerical instability. The matching results of the time history tests at ISMES (§5.5.3.2) and LNEC (§5.5.4.2) came from the use of a combination of transfer function updating and non-linear iterative matching.

Calculation of system transfer functions – Several of the techniques described above rely on the calculation of various transfer functions for the shaking table system. These calculations can vary greatly in complexity. In the simplest case a single transfer function is calculated for the drive and acquired signals in each table axis, resulting in 6 transfer functions for a six-axis table. At the start of this research programme, this was the system used at the four laboratories. This allowed iterative matching of any motion on the shaking table, but did not directly compensate for any cross-coupling between axes. A solution to this was the calculation of the transfer functions for all cross-coupling terms as well as the direct transfer functions, resulting in a matrix of 36 transfer functions for a six-axis table. This transfer function matrix could then be used to control cross-coupling between table axes when generating new drive signals as part of the iterative matching process. A further level of sophistication, now in use at ISMES, is the use of additional transducers to provide extra transfer functions between drive and acquired motions. This results in a non-square matrix of transfer functions where there are more acquired signals than drive signals (72 if accelerometers and displacement transducers are used to record the motion of each table axis). The difficulty then is the inversion of this non-square matrix so that the inverse transfer functions can be used to generate new drive signals. Fortunately this matrix inversion is possible if the higher frequency components of the inverse transfer functions are calculated from the acceleration data while the lower frequency components of the inverse transfer functions are calculated from the displacement data. The cut-off frequency for the change between the two types of instruments will need to be chosen based on the types of instruments being used. The form of the weighting function that is used to combine values from both types of instrument around this frequency will also need to be based on the types of instruments used and experience gained during previous matching. By using accelerometers and displacement transducers to monitor the platform motion and then combining these transfer functions over different frequency ranges as the matrix is inverted, a much more accurate inverse transfer function matrix can be calculated. The only disadvantages of using this extended transfer function matrix are that all the instruments must be accurately and consistently calibrated, and that it will take twice as long to calculate all the transfer functions after each iteration of the matching process.

Number of matching iterations required – It should be noted while the non-linear iterative matching algorithm can produce the best platform motion, it can take many iterations to do so. For example, in ISMES the excellent matches with the flexible specimen (e.g. figure 5.34) were only achieved after 18 iterations. The linear matching algorithm is much quicker, and in most cases a good match was achieved after 2 or 3 iterations (e.g. figure 5.12). The decision as to the type of iterative matching algorithm (linear or non-linear) that will be employed for a particular test will therefore depend not only on the accuracy of motion required, but also on an assessment of the ability of the model being tested to withstand a significant number of pre-test shakes during the matching process. In the case of a linear model, i.e. a model that will not experience any damage during a shaking table test with the desired motion, it will be acceptable to shake the model repeatedly during the matching process, as the final test results will not be affected by these pre-tests. In this case the platform motion can be matched as accurately as possible using the non-linear iterative matching techniques. Unfortunately, it is very unusual for specimens being tested on shaking tables for earthquake engineering research purposes to be linear at the levels of shaking normally applied. However, the tests performed using the flexible specimen (figure 5.6) on the four ECOEST shaking tables could be performed using both linear and non-linear iterative matching techniques because the specimen was designed to remain linear under any reasonable loading.

If a non-linear model is to be tested, i.e. a model that will experience significant damage or will collapse during a shaking table test with the desired motion, then the user is faced with a choice between several options, none of which is perfect. However, any of these techniques, when used in the correct circumstances, will result in a good experimental test.

1. Iteratively match the time history accurately at full scale without the specimen on the platform, then mount the specimen and test it with the matched earthquake. Additional errors in the earthquake reproduction will be caused by the new table-specimen interaction caused by the change from the bare platform to a system with the platform plus specimen. Unless the specimen is very small compared to the shaking table platform (from experience, less than about 10% of the mass of the platform) the errors introduced by the change in table-specimen interaction are likely significantly to affect the accuracy of the earthquake reproduction. If the specimen is small, either linear or non-linear iterative matching will be appropriate as the platform motion will not be significantly affected by the addition of the specimen. However, if any significant table-specimen interaction is expected (because the specimen has a high mass), there is no point spending a lot of time performing an accurate non-linear match of the time history which will subsequently be changed radically by the table-specimen interaction. In this case, performing a linear match, mounting the specimen, and performing the actual test will result in platform motions that are likely to be just as accurate, in significantly less time. It should be noted, however, that the final platform motions are unlikely to be very accurate if a large specimen was mounted on the platform, so unless the specimen to be tested is very small compared with the platform mass (less than about 10% of the mass of the platform), this method is not recommended.

2. Iteratively match the time history at low amplitudes with the specimen on the platform then scale up the drive signals for the actual test. Additional errors in the earthquake reproduction will then be caused by a change in table-specimen interaction based on the fact that the specimen was behaving linearly during the matching but not during the actual test. There may also be further errors resulting from any non-linearity in the amplitude response of the shaking table. The errors resulting from this process are likely to be less severe than in the previous case, so either linear or non-linear iterative

matching will be appropriate. However, if it is known that the table does not respond linearly to a change in amplitude of the drive signal, then this method will not be appropriate at all. This is likely to be the case when the earthquake motion to be matched is close to any of the physical limits (acceleration, velocity or displacement) of the table. At the extremes of performance any table is least likely to behave as a linear system, so the scaling of drive signals is least likely to work.

3. Mount the specimen on the platform, but restrain it as much as possible so that it cannot move relative to the platform (normally accomplished by adding extra bracing or by fixing the specimen rigidly to the support frame(s)) and iteratively match the time history at full-scale. The benefit of this approach is that the overturning moment due to the high centre of gravity of the specimen is compensated for in the matching process. The errors, in this case, again come from a change in table-specimen interaction, based on the fact that the specimen was behaving rigidly during the matching but not during the actual test. This is the most common technique used to test non-linear specimens on a shaking table, and either linear or non-linear iterative matching of the platform motions will be appropriate. However, in this and in the second case, it may still be unacceptable to run too many pre-tests as part of the matching process as some degradation of the model may occur, and the shaking table user must therefore make a judgement as to whether the accuracy of platform motion is more important than the integrity of the specimen for the final tests.

6.3.2 Development of new time history matching software

Currently a major difference between the four sites is the use of completely different computer systems to control the shaking tables. The differences in software used at each site are mainly a result of this different hardware. Therefore, before any common software could be developed at the four sites, there had to be some standardisation of the computer hardware used. This is now beginning to take place as the MCS (Minimal Control Synthesis) algorithm (Stoten, 1993) is being implemented at the four laboratories (§6.7). This software runs on a PC within the Windows environment, and with relatively minor modifications the same software is now being used to control all four shaking tables. This software also has the potential of controlling shaking tables in real-time, avoiding the problems associated with the iterative matching techniques described above (§6.3.1).

However, it may be some time before MCS is implemented completely at the four sites, and extensive testing (such as that performed as part of this research) still has to be performed to prove the robustness of the MCS system in the control of shaking tables. Until this happens the iterative matching techniques (§6.3.1) will continue to be the most reliable methods for controlling platform motions.

Currently the time history matching software in Bristol (MATCH ver. 1.0) can only match displacements or accelerations, and cannot perform an iterative matching process for both at the same time. In addition to this problem, the matching software does not take any cross-coupling of axis motions into account. The software used in Athens to control the iterative matching process uses similar procedures to those used in Bristol, but because the Athens table can be tuned more accurately the software does not have to compensate for the large resonance effects present in the Bristol table. The tests at ISMES and LNEC using non-square transfer function matrices, which allowed measurements from accelerometers and displacement transducers to be used in the matching process, produced some excellent results even with the flexible specimen mounted on the platform (§5.5.3.2 and §5.5.4.2). Equivalent software is currently being developed at Bristol to improve the iterative matching processes at Bristol further.

6.4 Active control of passive test axes

One of the most important findings of this research was the necessity of actively controlling (with software) all shaking table axes, including those that should experience no movement during any particular shake. Before this research started, the shaking table operators at the four ECOEST laboratories would use the various iterative software matching techniques to recreate specific motions on specific table axes. However, the software was generally not used to try to force a zero motion into the axes that were not to be excited. For example, if a single-axis reproduction in the X axis of the El Centro shake was required, the matching software was used to perform the iterative matching based on the desired and achieved motions of this axis only. Any motion that occurred in the roll axis resulting from cross-coupling of the axis motions or from table-specimen interaction would then be controlled only by the hardware control system. The frequency response tests at the four laboratories (Bristol §5.5.1.1, Athens §5.5.2.1, ISMES §5.5.3.1 and LNEC

§5.5.4.1) showed the potential dangers in this approach. The tests at all the laboratories showed that even the bare tables displayed some cross-coupling between axis motions, and this cross-coupling became more severe with the flexible specimen on the platform. The effect of this lack of control over the axes not directly excited can be seen in figures 6.4 and 6.5, which show the rotational accelerations and displacements occurring on the ISMES table for a three-axis match of the Kalamata earthquake with an 8 tonne rigid payload. The rotational axes, in this case, are only controlled by the hardware control system, and both figures show interaction between the translational and rotational axes, particularly around 8 seconds into the shake where the translational motions are strongest. The lateral accelerations and displacements of the platform during the same shake are shown in figures 6.6 and 6.7. A very good match of platform accelerations has been achieved in the lateral axis, but the platform displacements are not so good. In this case the software was set up to match accelerations over a frequency range of 0.5 Hz to 35 Hz, but the good acceleration response was at the expense of a good displacement response (this problem is discussed further in §6.5). The software was then modified to control the accelerations in all six table axes, again over a frequency range of 0.5 Hz to 35 Hz. This resulted in a noticeable improvement in the rotation accelerations (figure 6.8), with the best improvements occurring in the Roll and Yaw axes. The lateral accelerations and displacements of the platform during this shake are shown in figures 6.10 and 6.11, and are no different to those achieved during the three-axis matching. However, the rotational displacements, which are now being controlled by the software, show some low frequency (< 0.5 Hz) motion (figure 6.9) which is significantly worse than that which occurred in the test using three-axis matching. This was caused by the frequency range over which the software was working, and any motion below 0.5 Hz, in this case, was effectively uncontrolled. In order to try to control this low frequency rocking of the platform the software was further modified to perform a six-axis match, but this time of the platform displacements over a frequency range of 0.1 Hz to 20 Hz. The resulting rotational accelerations and displacements are shown in figures 6.12 and 6.13. The rotational accelerations are slightly noisier than in the previous case, where the accelerations of the platform were directly controlled, but the rotational displacements are much better and almost negligible. In addition, the lateral displacements of the platform during this shake (figure 6.15) are much better than in the earlier tests and the lateral accelerations of the table (figure 6.14) are still good, although there is some loss of accuracy above 8 Hz in comparison with the accelerations achieved during either of the acceleration matches (compare figure 6.14 with figure 6.10).

These results raise several important issues:

- All shaking table axes should be actively controlled by the software, even if the desired motion in any axis is zero. This will improve the performance of the shaking table, minimise table-specimen interaction and result in more accurate specimen response (§5.5.3.2).
- The frequency range adopted during the processing of data as part of the iterative matching process is important, and should be as wide as possible while still producing a numerically stable algorithm. Generally this range can only be determined by experience. However, based on the results of this research, some guidelines for this range are: Upper value = maximum frequency component of the motion being matched; Lower value = lowest operating frequency of the accelerometers used to monitor the platform motion for acceleration matching (about 0.3 Hz), or approximately 0.1 Hz for displacement matching.
- Shaking tables (like the LNEC table) that are constrained by some mechanical system to move in only certain axes, are limited by the stiffness of the mechanical constraining system with regard to the accuracy of control in the constrained axes. If such a constraining system is very stiff, such tables will perform better than tables with all axes unconstrained. However, tables that can be actively controlled in all axes will perform much better than tables with relatively flexible constraining systems if all the axes of the unconstrained table are actively controlled.
- The shaking table user must be aware that currently even the best shaking table control system cannot perfectly match platform accelerations and displacements at the same time. The iterative matching process is always a compromise between a very accurate acceleration response at the expense of the displacement response or a very accurate displacement response at the expense of the acceleration response; this compromise is discussed further in §6.6.

6.5 Measurement of table movement

When the first series of iterative time history matching tests was performed at Bristol and Athens (§5.5.1.2 and §5.5.2.2) it was not appreciated how significant small rotations were on table performance, and how they might affect the behaviour of any specimens attached to the platform. Because of this oversight, instruments were not placed on the platform specifically to measure the rotational accelerations and displacements at these two sites. Therefore, although the results of the time history matching at these two sites were excellent in the translational axes, it is not known whether there were also significant rotational components in the platform motion that would have changed the effective force applied to the flexible specimens. In the tests in ISMES and LNEC (§5.5.3.2 and §5.5.4.2) this oversight was corrected, and the rotational components of the platform motion were monitored in addition to the translational motions.

Following this research programme, it is now considered that the measurement of all platform motions (preferably both accelerations and displacements) during any testing is essential for several reasons:

- Without measurement of all the platform motions, the extent of any errors in the matching process will be unknown.
- Without measurement of all the platform motions, the extent of any shaking table-specimen interaction will be unknown.
- If the recorded errors in platform motion are significant, then more sophisticated iterative software matching processes can be used (§6.3.1) to control and reduce the errors.
- If the errors in platform motion are recorded, then the effective forces applied to the specimen on the platform can be calculated taking into account these undesired motions. In this way the data from the tests can be adjusted to compensate for the actual behaviour of the shaking table. One method for performing this adjustment is described by Rinawi et al. (1988), and simply adds (or subtracts) the effective translational forces created by the rotations to (or from) the forces generated by the actual translational motions.

The main issue is that it is very unwise to assume that shaking tables are performing perfectly. Only by measuring the actual platform motions, including those motions that are expected to be zero, can we be certain that the test actually being performed is the same as we intended. This finding has implications for the quality of results from previous shaking table experiments around the world. However, it should be stressed that the flexible model used in this research programme was designed to test the limits of the four shaking tables, and proved to be a quite extreme test for the table control system at Bristol in particular. The vast majority of shaking table tests are not performed at the performance limits of the tables, and therefore table-specimen interaction is less likely to be a problem. If table-specimen interaction is noticed, then this research has shown that the iterative matching techniques described in §6.3.1, combined with the practice of actively controlling the motion in any passive test axes (§6.4), will result in excellent performance of a shaking table.

6.6 Acceleration or displacement matching

The difficulties of recreating accurate displacement and acceleration motions on a shaking table at the same time have already been highlighted in the results of the tests at the four laboratories. These difficulties are, in essence, caused inability to measure either displacement or acceleration signals accurately across the full operating range of a shaking table. Generally, the response of a specimen on a shaking table will be determined by the acceleration of the table platform which controls the forces applied to the structure. It would, therefore, appear that measurement and control of the platform accelerations would be the ideal situation. Unfortunately, the types of robust accelerometers needed to record the high accelerations that can be achieved on a shaking table do not operate well at low frequencies. Therefore any iterative time history matching algorithm that tries to record and control only the platform accelerations is unlikely accurately to reproduce the low frequency (0 Hz to 5 Hz) components of a time history. Displacement signals, on the other hand, provide a way good way to quantify the low frequency components (0 Hz to 10 Hz) of the desired or achieved platform motions. However, the displacements associated with high frequency components of an earthquake signal are very small. For example, production of a 1g acceleration at 1 Hz requires a platform displacement of ± 250 mm, but production of a 1g acceleration at 100 Hz only results in a platform displacement of ± 24 μm. If such small displacements could be resolved while still measuring the large motions, then an iterative time history matching algorithm that tried to record and control just the platform displacements could, in theory, accurately control the table at all frequencies. Unfortunately, displacement transducers that can accurately resolve micrometers over a range of several hundred millimetres and operate at high velocities are rare, expensive, and cannot easily be incorporated into shaking tables. An additional difficulty results from the quantisation of either the acceleration or displacement signals by the acquisition system for subsequent analysis by the control software. The signal from a displacement transducer that could measure micrometers over a range of ± 150 mm (the actuator stroke of the Bristol table) would need to be acquired by a 19 or 20 bit A/D (Analogue to Digital) system. While such systems do exist, they are very expensive and may still not be accurate enough to resolve the displacements resulting from small high-frequency accelerations. Similar problems of quantisation occur when trying to measure the wide range of accelerations that result from low-frequency and high-frequency platform motions. One possible way to overcome these resolution and quantisation problems is the development of composite filters which can combine displacement and acceleration signals to form an equivalent, very high resolution displacement signal. These digital filters are being developed at Bristol, and preliminary results indicate that they may significantly improve our ability to measure, and hence control, platform responses over a broad frequency range.

All the current difficulties with recording the actual motion of a shaking table platform mean that, at the moment, no iterative matching algorithm will be able to control any table perfectly. However, the results of the tests at the four laboratories showed that the best combination of displacement and acceleration motions resulted from matching with displacement based drive and feedback signals. If the acceleration response of the platform is much more important than the platform displacements for a particular test, then matching with acceleration based drive and feedback signals will produce a better platform accelerations at the expense of much less accurate platform displacements. The use of a non-square transfer function matrix (§6.3.1), to try to control both displacements and accelerations at the same time, improved the matching in both of the cases described above. However, matching with a displacement based drive signal using both sets of feedback signals still out-performed a match with an acceleration based drive signal which also used both sets of feedback signals.

6.7 Real-time control of shaking tables

Traditionally, to minimise the problem of table-specimen interaction, shaking tables have been controlled using off-line iterative approaches which can, in theory, produce the required platform motions if the combined system is linear. Unfortunately, in many cases the specimen being tested will have significantly non-linear dynamic characteristics, and will often experience some form of failure during testing. No off-line iterative process (§6.3) can compensate for these kinds of change in the dynamic characteristics of shaking table-specimen system, and more advanced control methodologies are therefore needed. Additional problems occur when trying to control the undesired motion of axes which are required to have zero motion (§6.4). For example, a heavy specimen may induce a coupling between horizontal and rotational motions. If these rotations are not limited then the response of the structure can be significantly affected (§5.5.3.2).

To try to solve these problems, a recent innovation in control technology is currently being implemented on the four ECOEST shaking tables. The MCS (Minimal Control Synthesis) algorithm effectively works by modifying the gains in all the feedback loops (§6.2.2) in real-time. This allows the compensation of any changing or unknown dynamic characteristics in the whole table-specimen system. The MCS algorithm has been used successfully in many robotics applications (Stoten, 1993), and it is hoped that, when fully implemented, it will remove the interaction between specimen and table.

There are two distinct types of control strategy that can be used in implementing MCS on a shaking table (Stoten and Gomez, 1998). The choice of strategy will depend on whether the algorithm can be incorporated directly into new control hardware, or whether MCS is used as a retrofit to improve the performance of existing control hardware. At Bristol, the replacement of the old analogue hardware control system with the new DARTEC 9600 digital controller gave the opportunity of incorporating the MCS algorithm into the control of each of the eight individual actuators. The strategy used at the other three sites has been to use MCS as an outer control loop to improve the performance of the existing control hardware. Both strategies have advantages and disadvantages that are discussed further by Stoten and Gomez (1998).

Initial experimental studies using the six-axis shaking table at Bristol, and a single-axis table and the six-axis shaking table at ISMES, have shown that an excellent match of the required displacement and acceleration time histories can be achieved once the MCS control algorithm is used to control a shaking table. These results can be achieved even if there are gross parameter changes in the table-specimen dynamics.

The following preliminary results (taken from Stoten and Gomez, 1998) show the extent of the improvements that can be achieved using MCS. The top plot of figure 6.16 shows the desired displacement of the single-axis table at ISMES. The middle and bottom plots (of figure 6.16) respectively show the error between the desired and achieved platform motions without and with MCS running. It should be noted that even through the hardware controller for this table had been very carefully tuned before these tests, there was a significant error between the desired and achieved motions with just the hardware controller. MCS significantly improved this performance. The desired acceleration, acceleration error without MCS, and acceleration error with MCS for the same test are shown in figure 6.17. Again MCS produced a noticeable improvement in platform response. The corresponding acceleration spectra achieved with and without MCS are shown in figure 6.18. The lines labelled Pd2xm are the spectra for the desired platform motions. These results show that MCS can significantly improve shaking table performance, but the real advantage of MCS is its ability to control a shaking table accurately even if there are real-time changes in the dynamic characteristics of the table. The results shown in figure 6.19 show the displacement errors that occurred when the tuning of the analogue controller was dramatically changed at a point 11 seconds into the shake. This could be considered to be equivalent to the total collapse a specimen on the table, which would dramatically change the dynamic characteristics of the whole system. Without MCS, the error between desired and achieved motions becomes as large at the desired motion itself. With MCS running, the system adapts to the changes and the displacement error remains small. These are the first known positive results from the implementation of any real-time control system on any shaking table, and they give an indication of what may be achieved by future shaking table control systems. Similar results to these have also been achieved on the six-axis tables at ISMES and at Bristol.

6.8 Conclusions and looking to the future

The results of the comparison of the four shaking tables (§5.5) highlighted the fact that all four shaking tables could accurately reproduce various desired platform motions. This was possible even though the four tables had significantly different performances. Significant linear table-specimen interaction could also be controlled, provided the operator was aware of the behaviour of the table and understood how the different methods of iterative earthquake matching software were best implemented (§6.3). The tests performed to date have shown that, as long as the shaking table-specimen system remains linear, the existing control algorithms, and those techniques developed during this testing, perform very well. A non-linear specimen is currently being developed that will allow the table control systems to be tested repetitively under more severe load conditions, and allow the more complicated control systems to be developed even further. It is hoped that eventually MCS will allow accurate real-time control such that the drive signal is continuously adjusted, keeping to the required platform motion even if the specimen exhibits significantly non-linear behaviour.

Looking to the future, it is difficult to predict how shaking tables will be used in five, let alone ten, years' time. The development of new technologies, and in particular the development of very high speed computers at low prices, means that many new possibilities for the use of shaking tables may soon become a reality. In 1995, a very unusual research programme involving the simultaneous use of three shaking tables, to test a 30 m bridge model weighing 50 tonnes, was performed in ISMES. The test required a great deal of planning (Casirati & Franchioni, 1994) and produced some very interesting results (Bousias et al., 1996). However, during testing, it was found that the simultaneous control of multiple shaking tables is a very difficult problem. The difficulties were cause by interaction of the specimen with the tables, and by interaction between the tables themselves as they each tried to excite the structure. In essence, a severe case of table-specimen interaction had to be controlled during this test. While this test was successful, because the errors were relatively small and because the actual platform motions were recorded for analysis later, there is obviously potential for further development of this type of testing in the future.

A further development in shaking table testing that may soon become a reality is the development of substructuring techniques that can be used in real-time on a shaking table. Shaking tables operate in real-time, therefore the models being tested will experience accurate inertial loading and any strain-rate effects will be modelled. However, they are limited by their relatively small capacity, and can only test small-scale models. On the other hand, pseudodynamic testing with sub-structuring allows full-sized structures or parts of structures to be tested, but not in real-time. A major development in real-time testing would therefore be the implementation of these types of substructuring techniques on shaking tables. This would allow experimental investigation into many new areas of structural dynamics, and in particular into some of the recent theoretical developments in soil-structure interaction. By using substructuring techniques on a shaking table, potentially it would be possible experimentally to verify theories that until now have been impossible to test because of the large volumes of soils required. The results of the initial testing of the MCS algorithm on the shaking table at ISMES and Bristol has meant that we can now contemplate performing real-time dynamic substructuring on a shaking table. This is because MCS has been shown to provide accurate control of shaking table motion in real-time, and can compensate for table-specimen interaction. Then, by modelling the soil analytically and coupling it with a physical structural model on the shaking table, large structural systems incorporating soil-structure interaction can theoretically be modelled.

Shaking tables are currently an essential tool for earthquake engineering researchers. Their performance, if used properly, is excellent, and in the future we are likely to see them being used in ever more innovative ways.

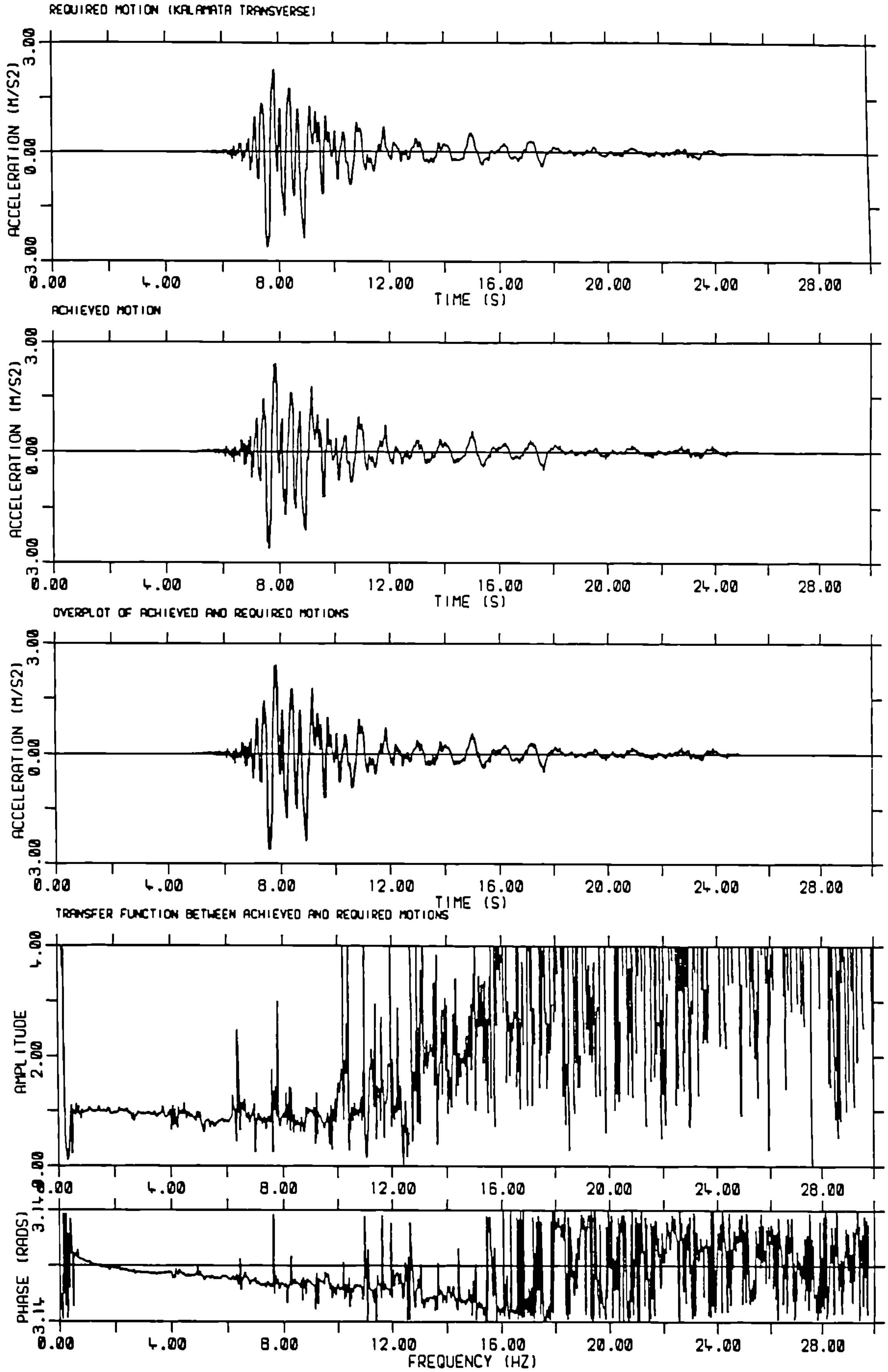


Fig. 6.1 Acceleration time history achieved on the ISMES table, on first iteration of the Kalamata shake, with no software precompensation

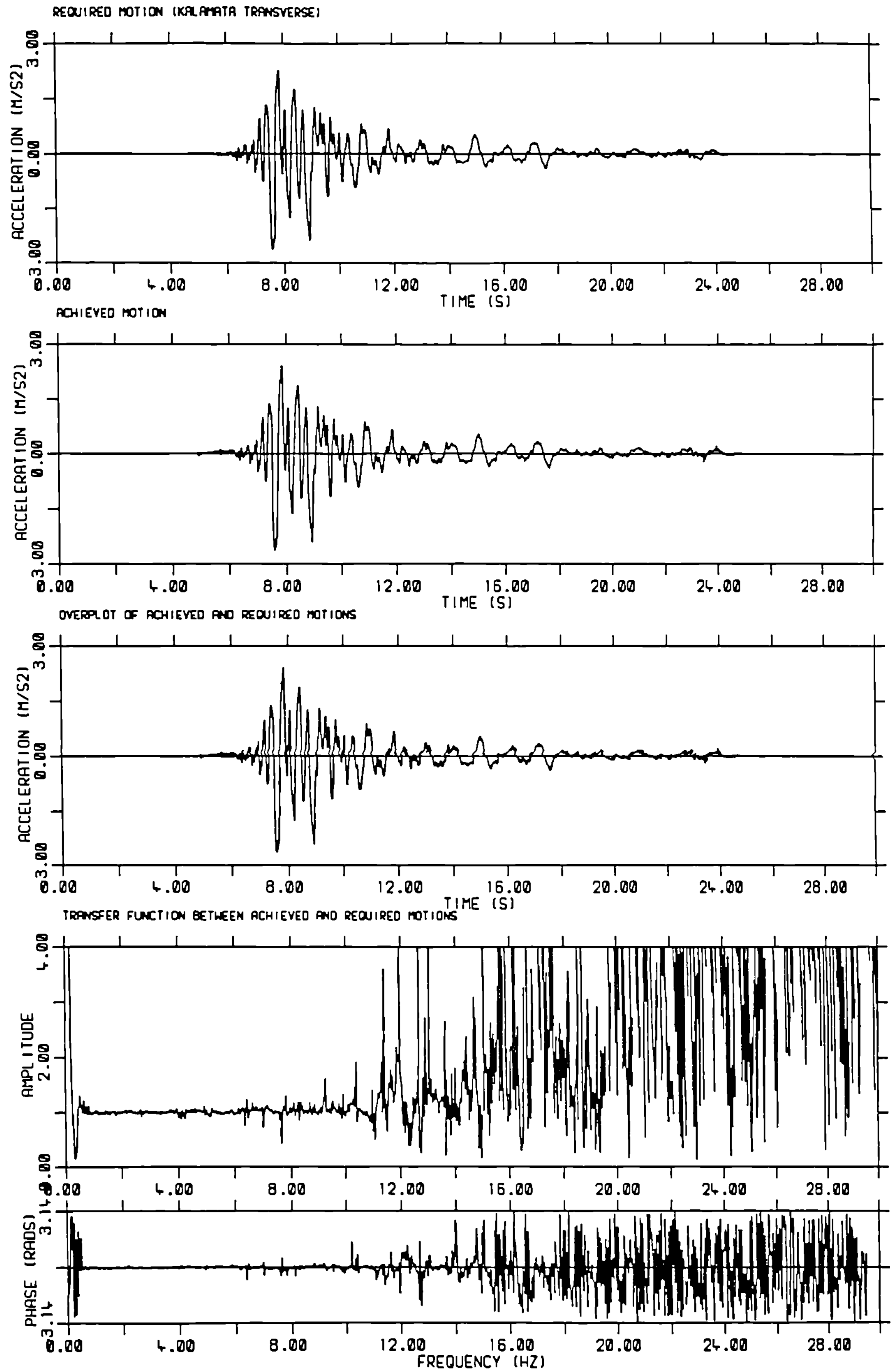


Fig. 6.2 Acceleration time history achieved on the ISMES table, on first iteration of the Kalamata shake, with software precompensation

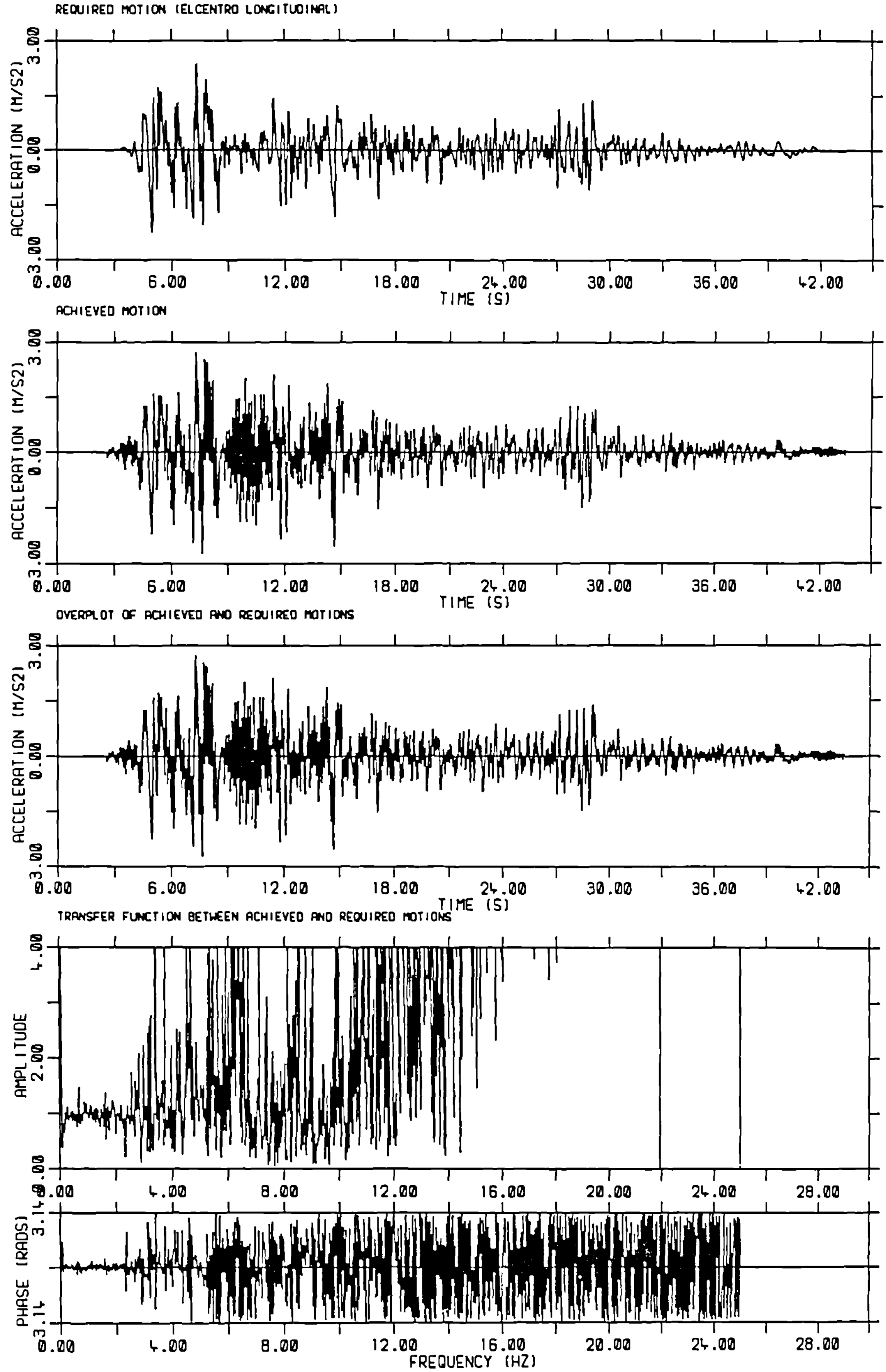


Fig. 6.3 Acceleration time history achieved on the Bristol table for a displacement match of the El Centro shake with no payload

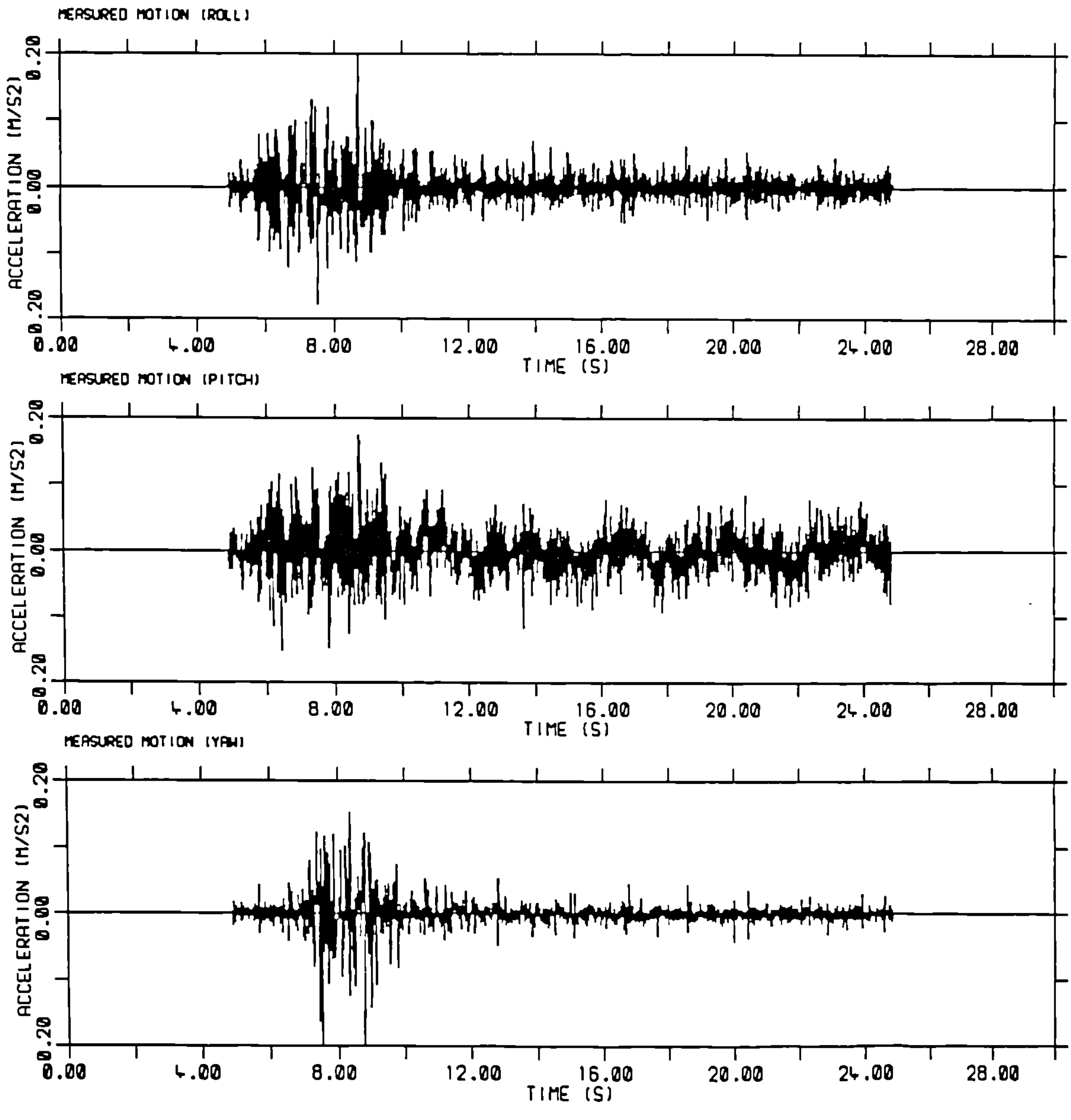


Fig. 6.4 Rotational accelerations of the ISMES table for a 3 DOF acceleration match of the Kalamata shake with an 8 tonne rigid payload

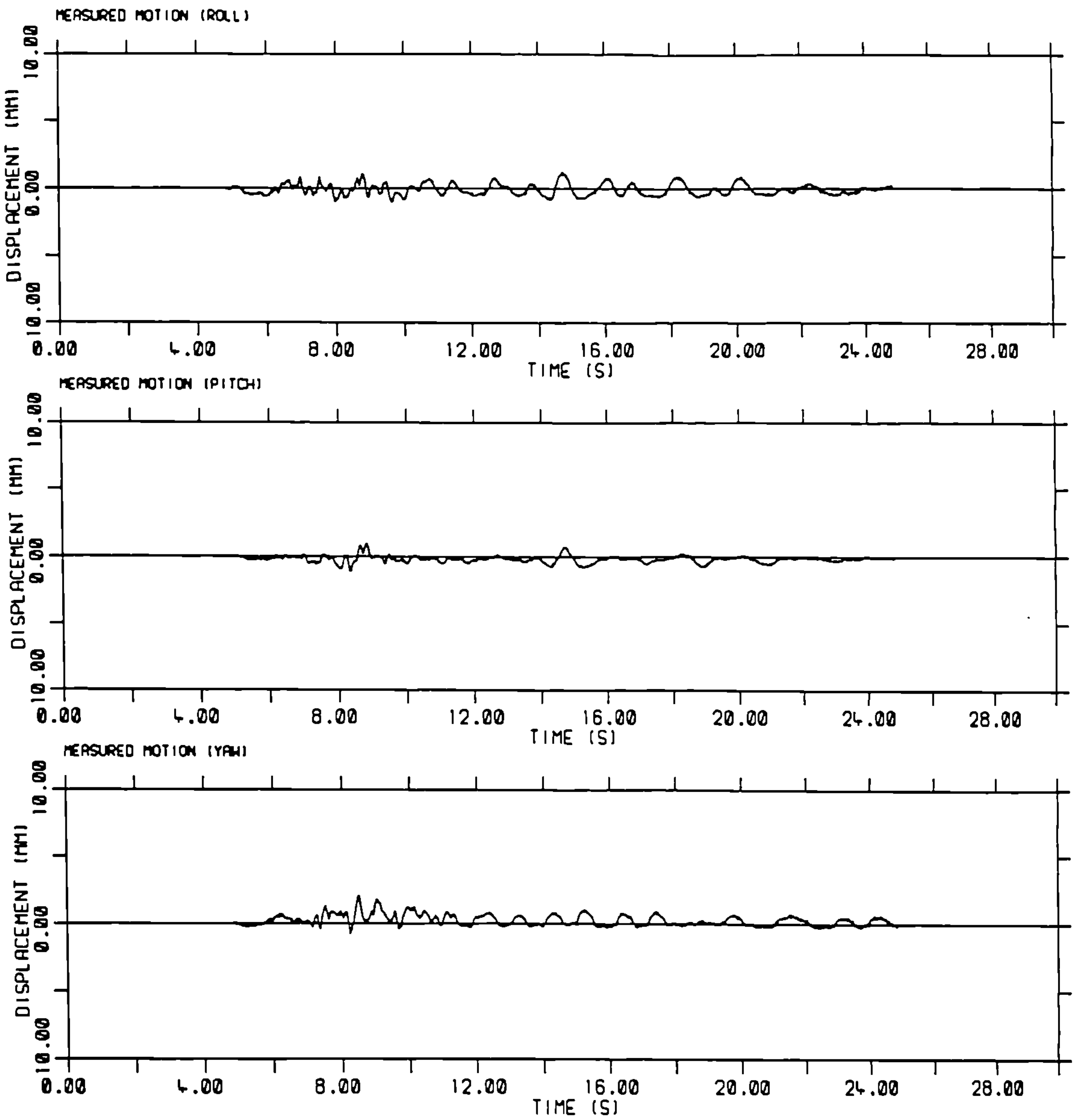


Fig. 6.5 Rotational displacements of the ISMES table for a 3 DOF acceleration match of the Kalamata shake with the 8 tonne rigid payload

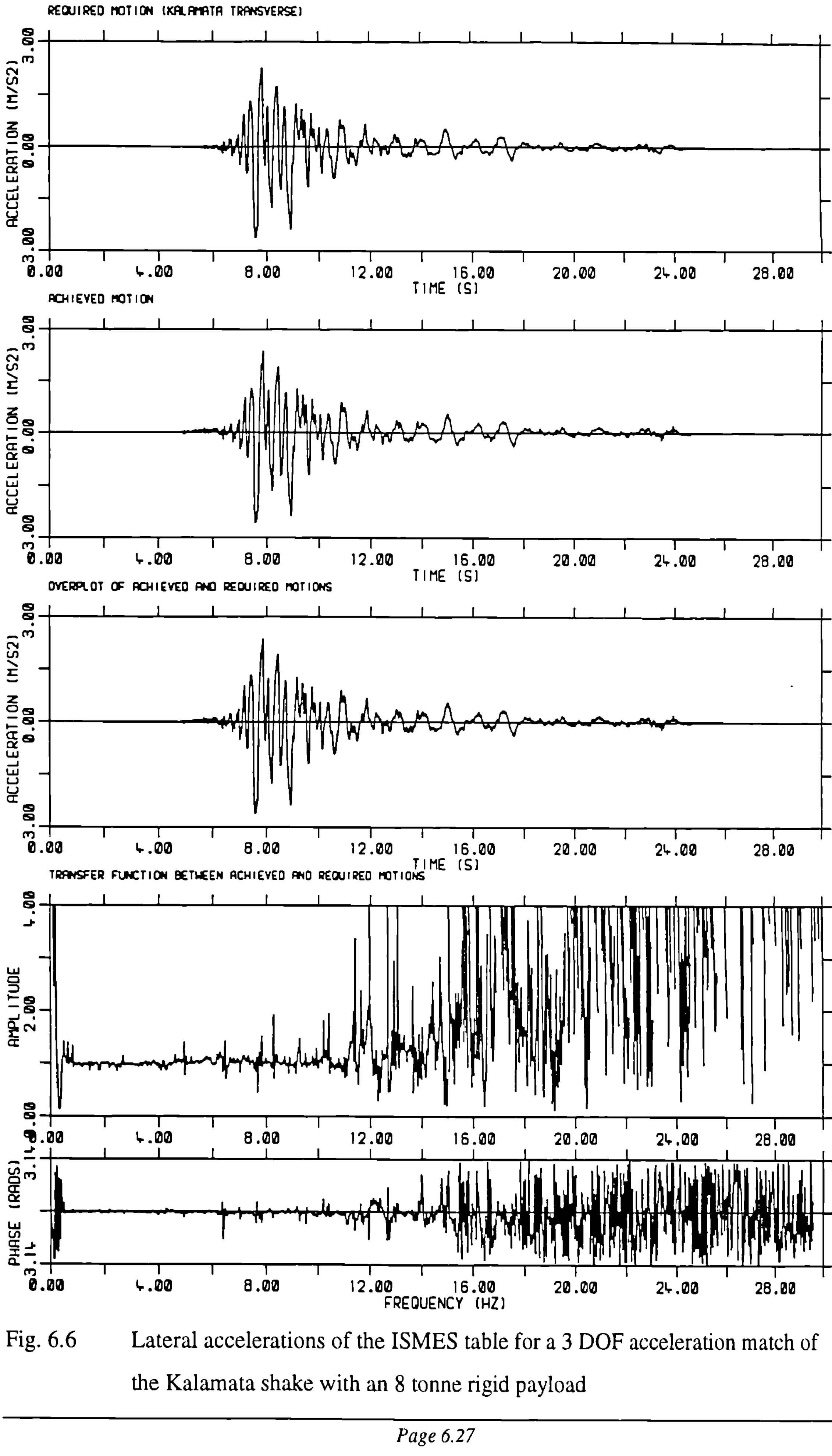


Fig. 6.6 Lateral accelerations of the ISMES table for a 3 DOF acceleration match of the Kalamata shake with an 8 tonne rigid payload

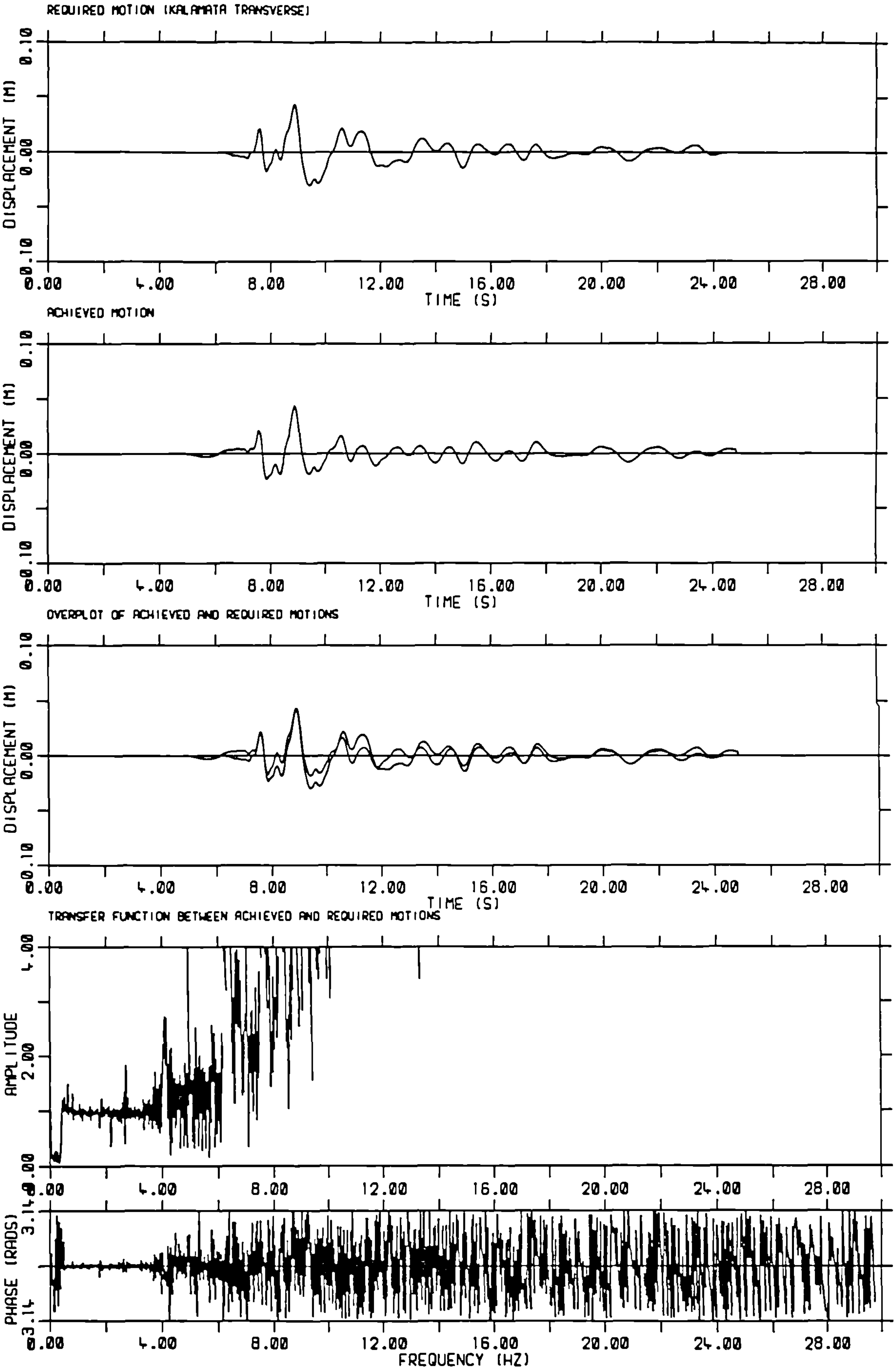


Fig. 6.7 Lateral displacements of the ISMES table for a 3 DOF acceleration match of the Kalamata shake with the 8 tonne rigid payload

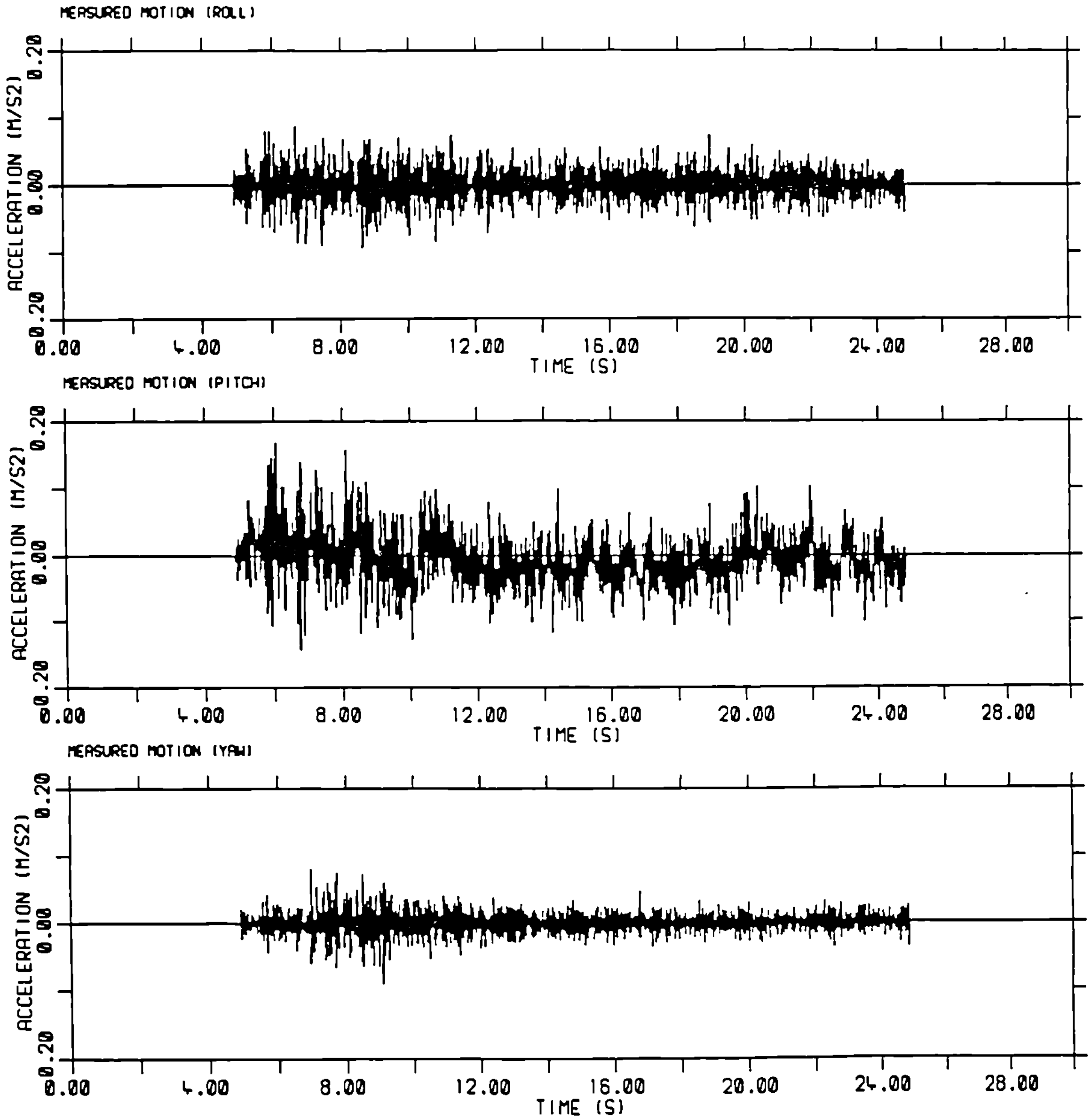


Fig. 6.8 Rotational accelerations of the ISMES table for a 6 DOF acceleration match of the Kalamata shake with the 8 tonne rigid payload

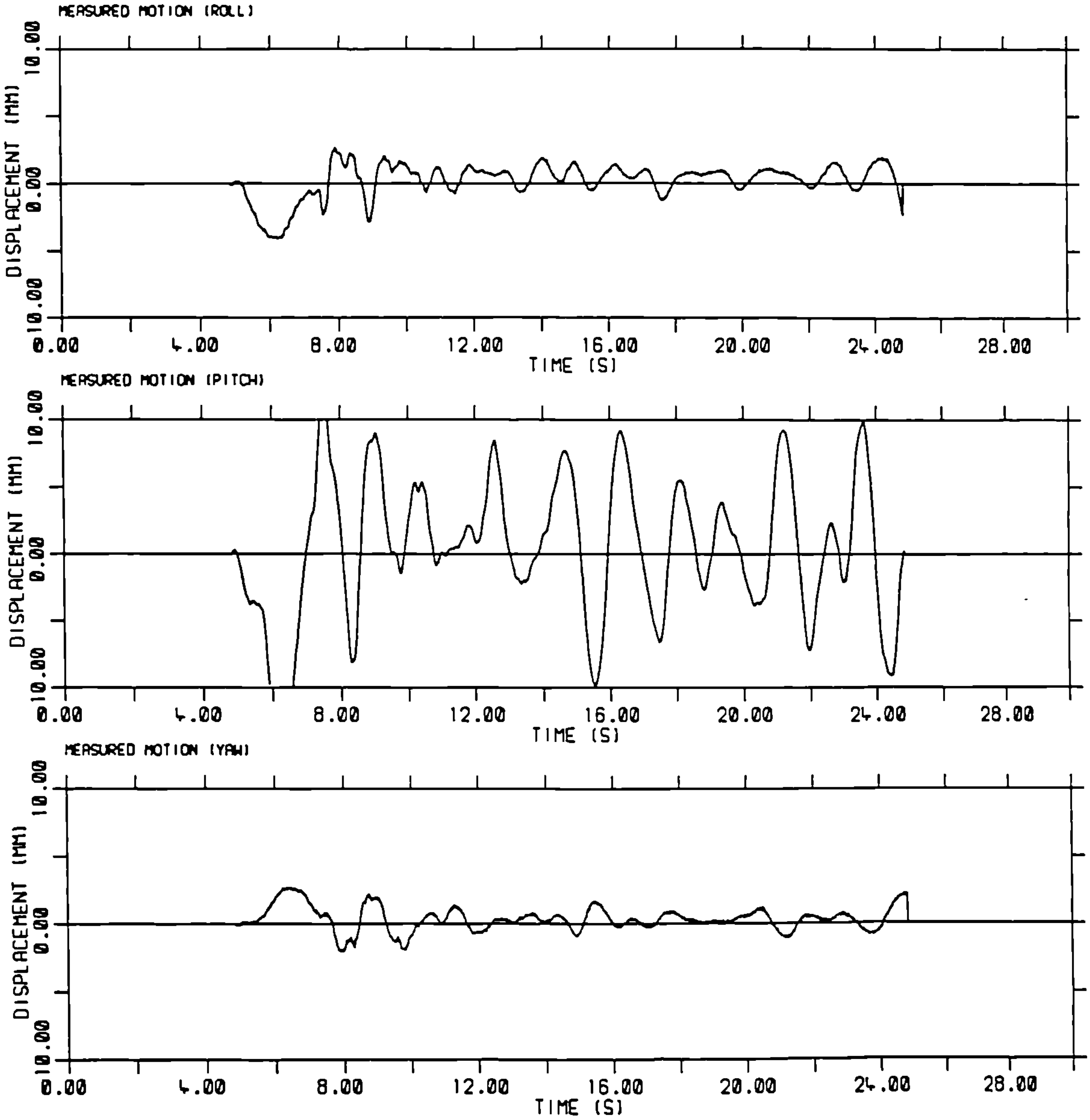


Fig. 6.9 Rotational displacements of the ISMES table for a 6 DOF acceleration match of the Kalamata shake with the 8 tonne rigid payload

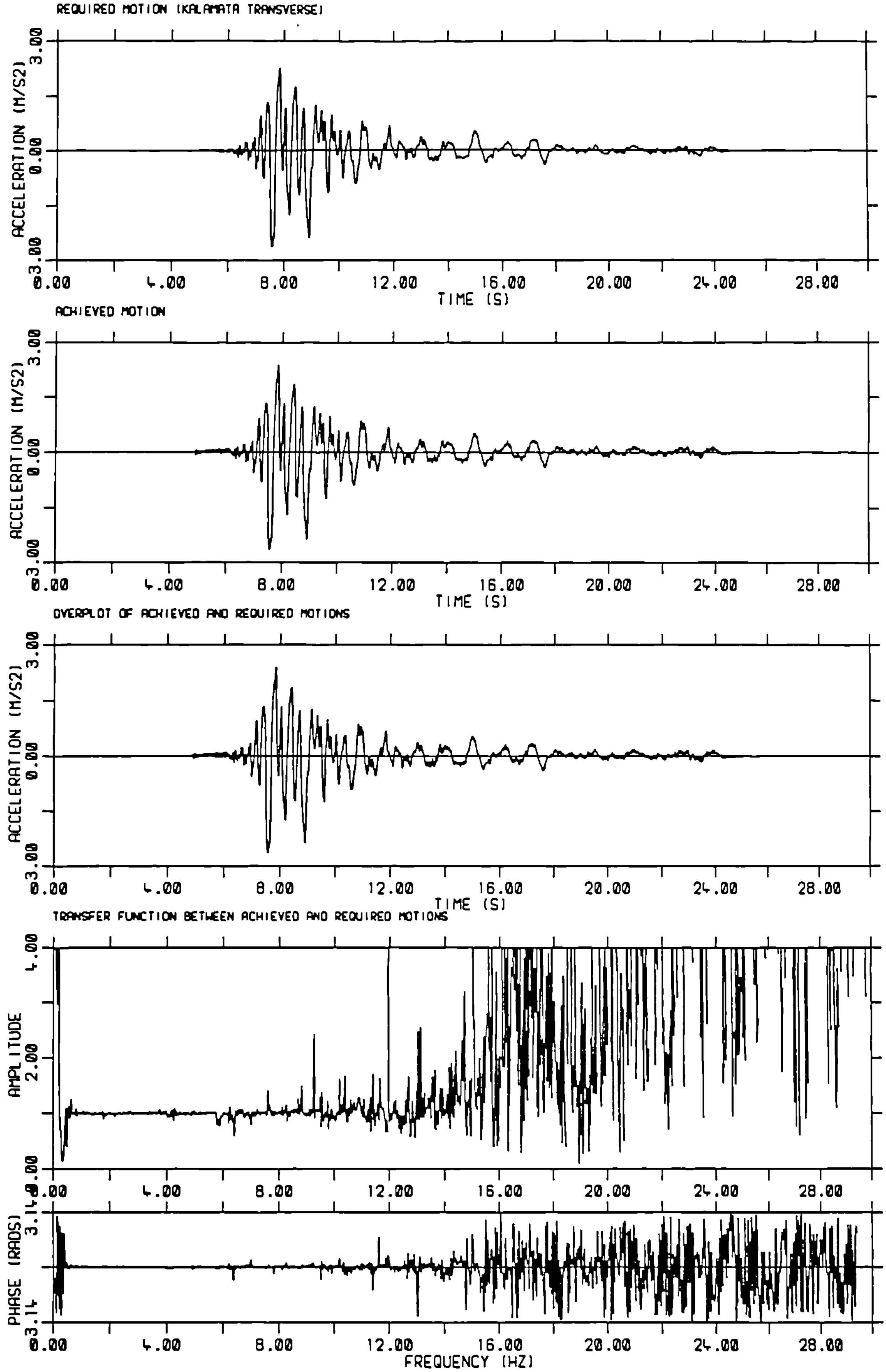


Fig. 6.10 Lateral accelerations of the ISMES table for a 6 DOF acceleration match of the Kalamata shake with the 8 tonne rigid payload

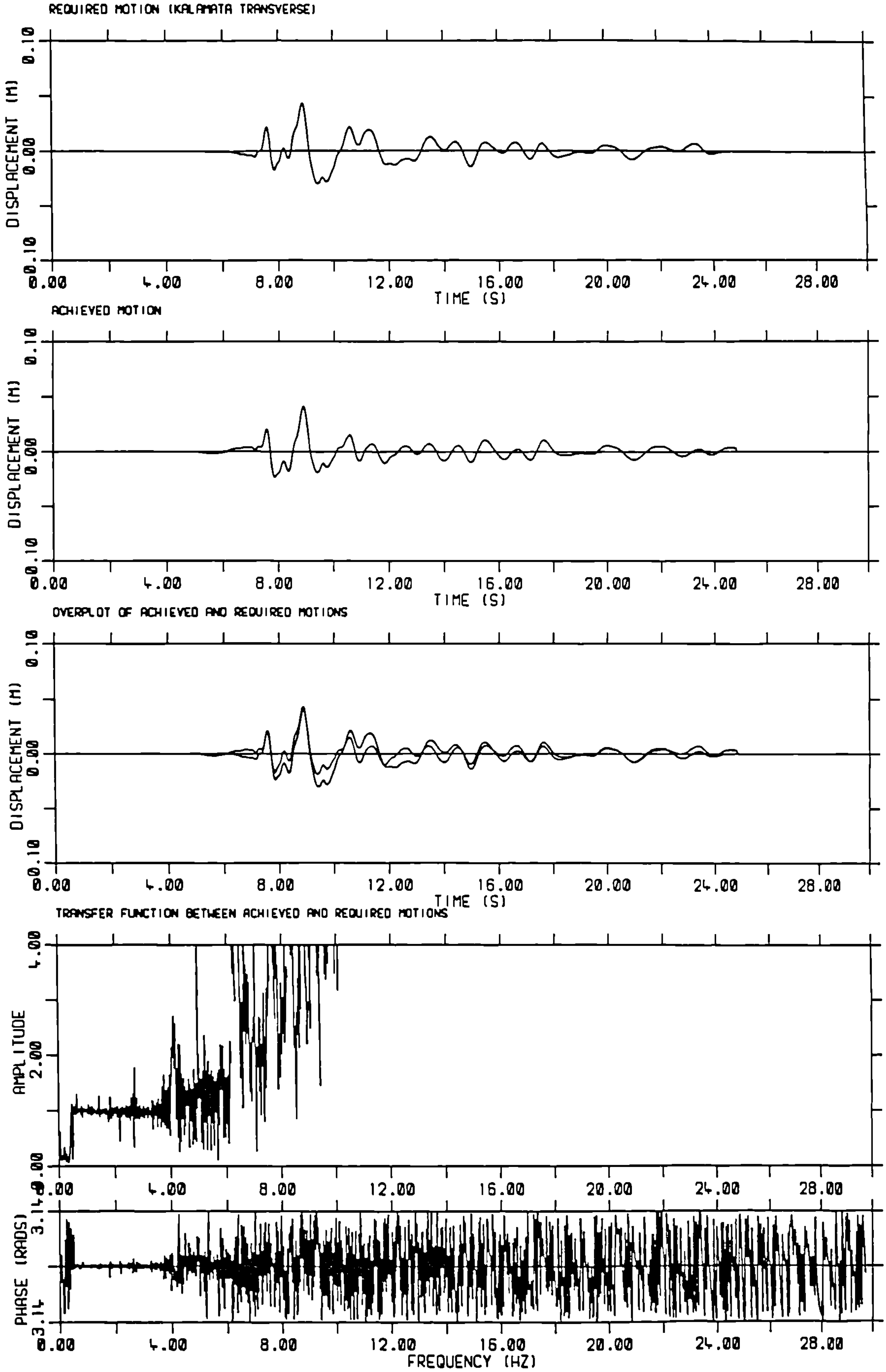


Fig. 6.11 Lateral displacements of the ISMES table for a 6 DOF acceleration match of the Kalamata shake with the 8 tonne rigid payload

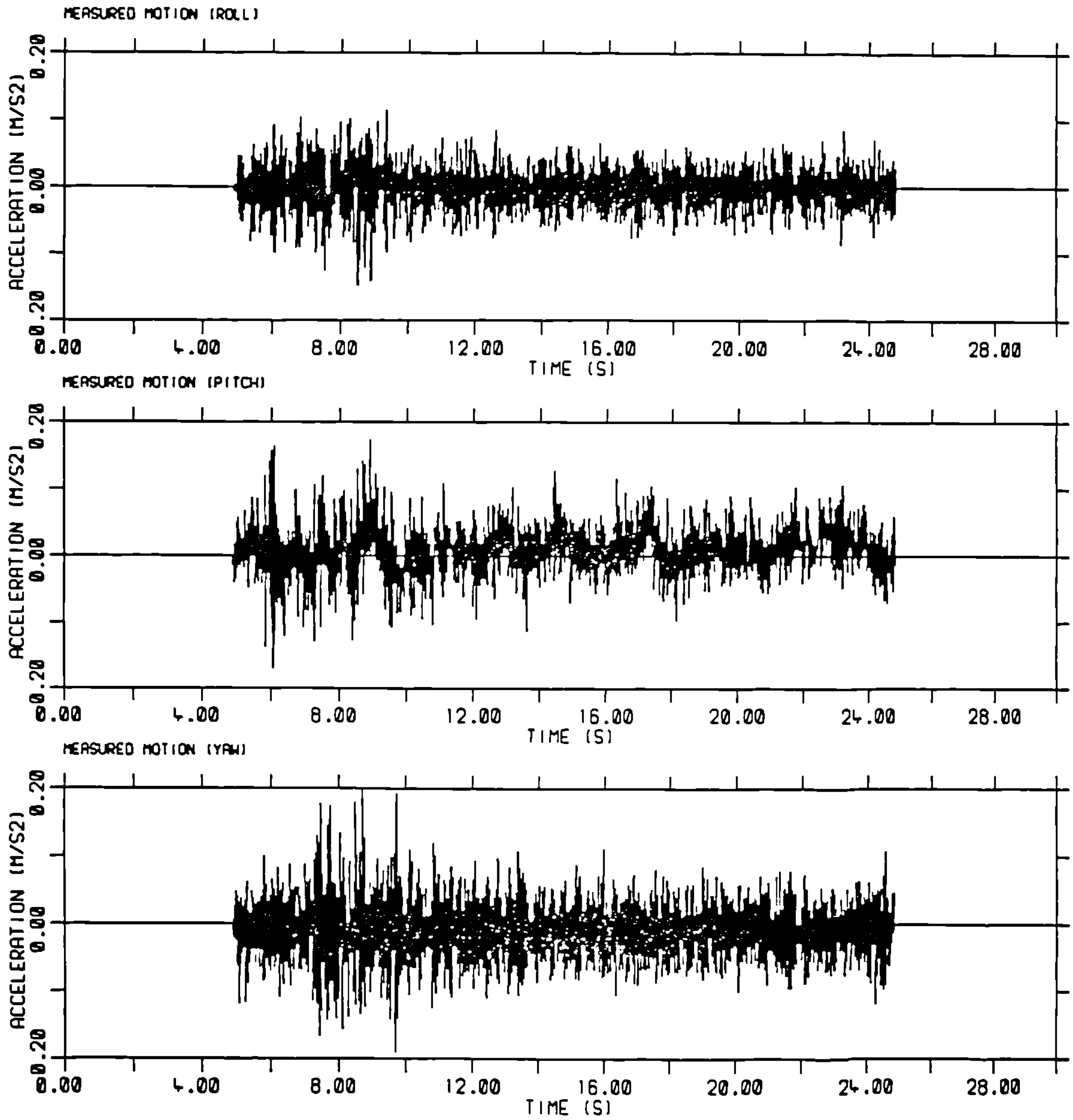


Fig. 6.12 Rotational accelerations of the ISMES table for a 6 DOF displacement match of the Kalamata shake with the 8 tonne rigid payload

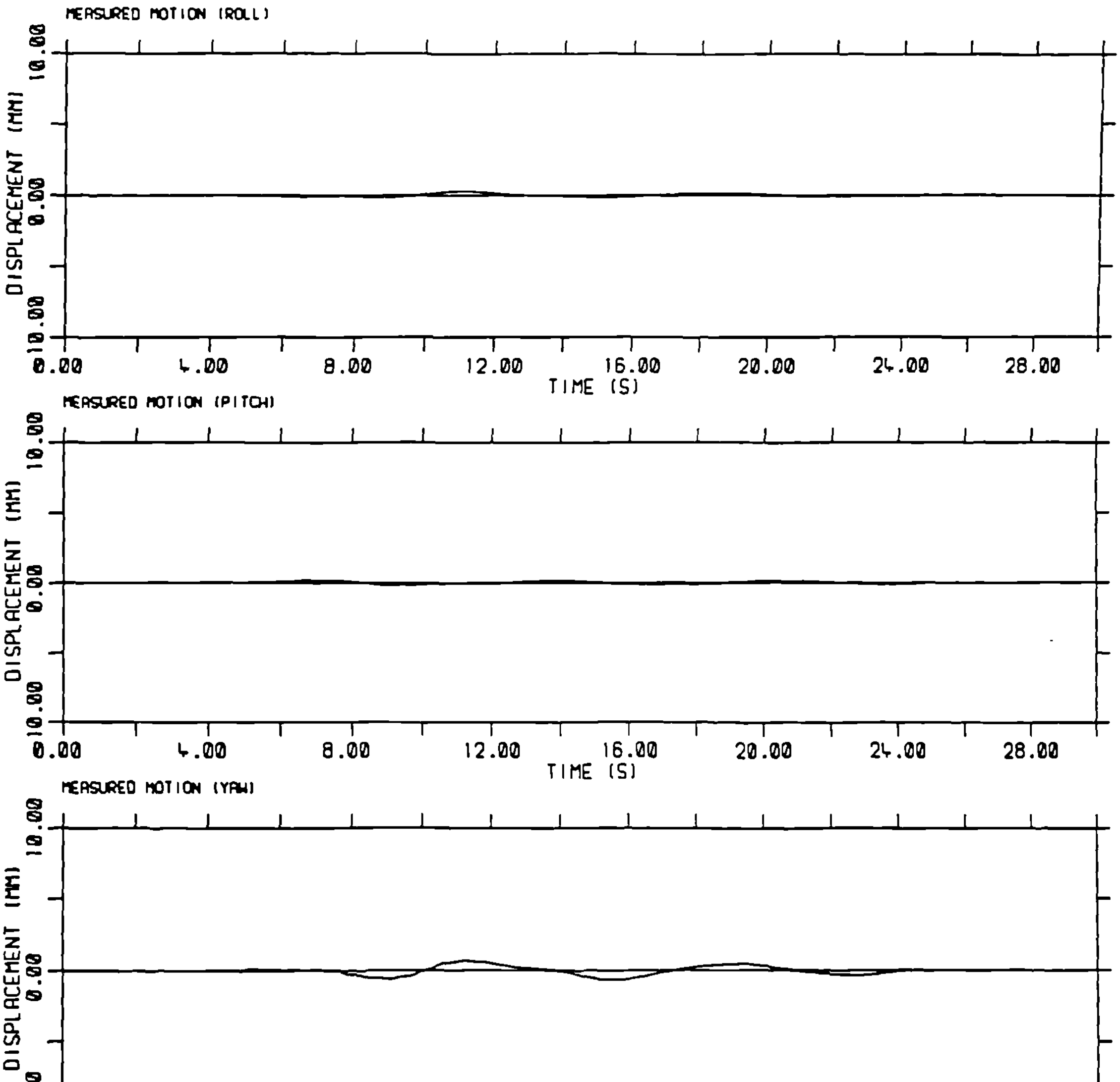


Fig. 6.13 Rotational displacements of the ISMES table for a 6 DOF displacement match of the Kalamata shake with the 8 tonne rigid payload

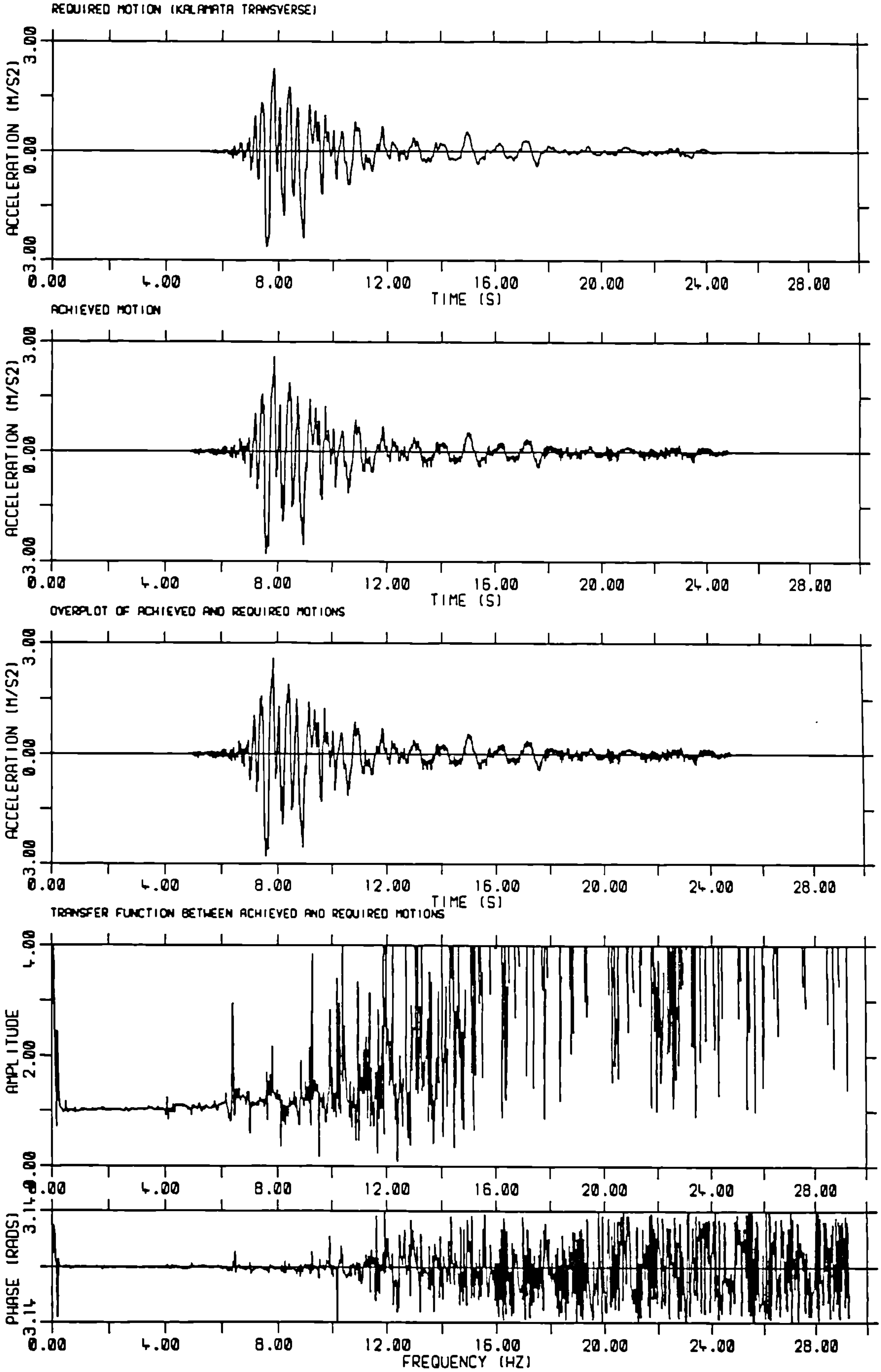


Fig. 6.14 Lateral accelerations of the ISMES table for a 6 DOF displacement match of the Kalamata shake with the 8 tonne rigid payload

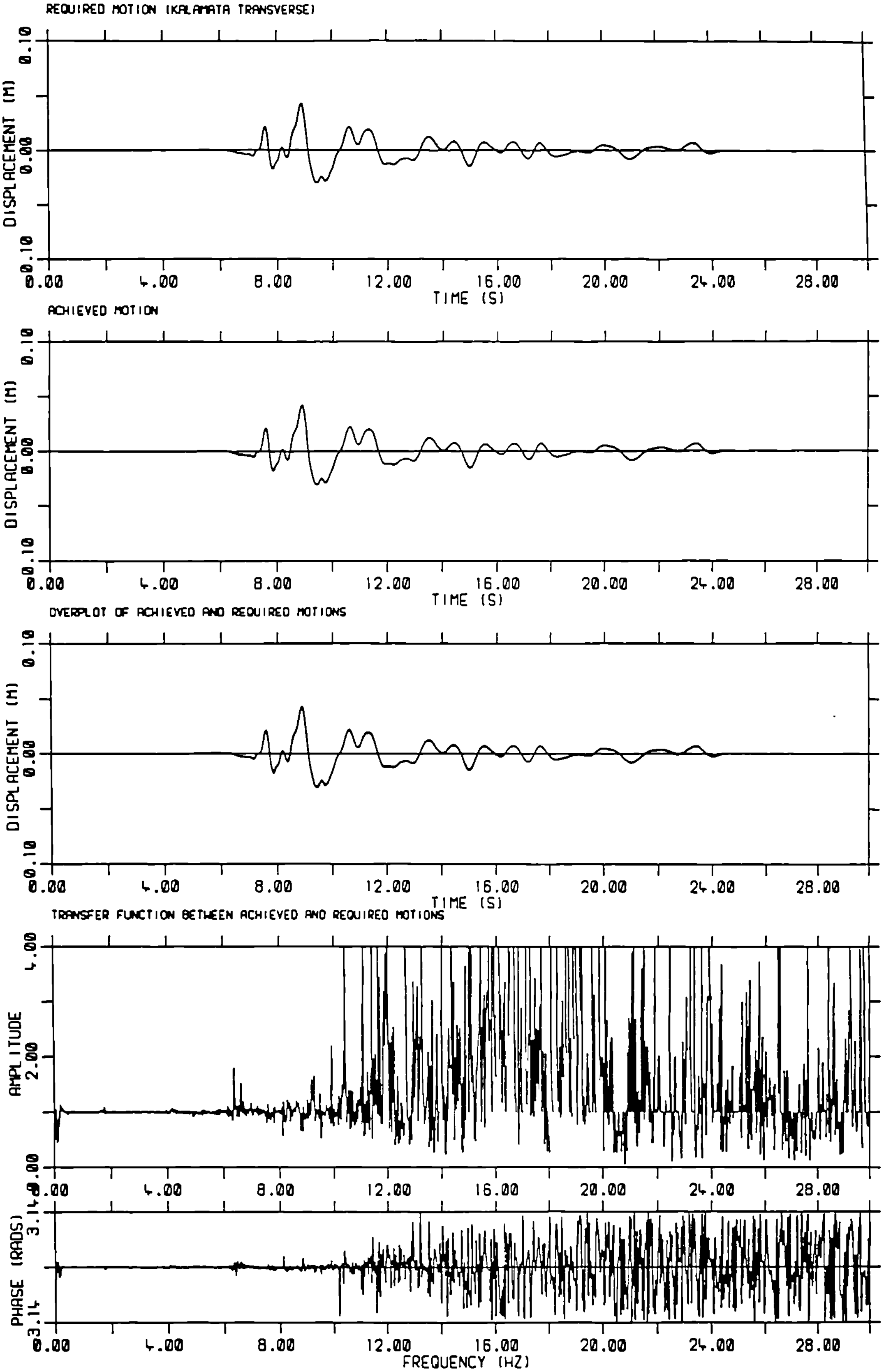


Fig. 6.15 Lateral displacements of the ISMES table for a 6 DOF displacement match of the Kalamata shake with the 8 tonne rigid payload

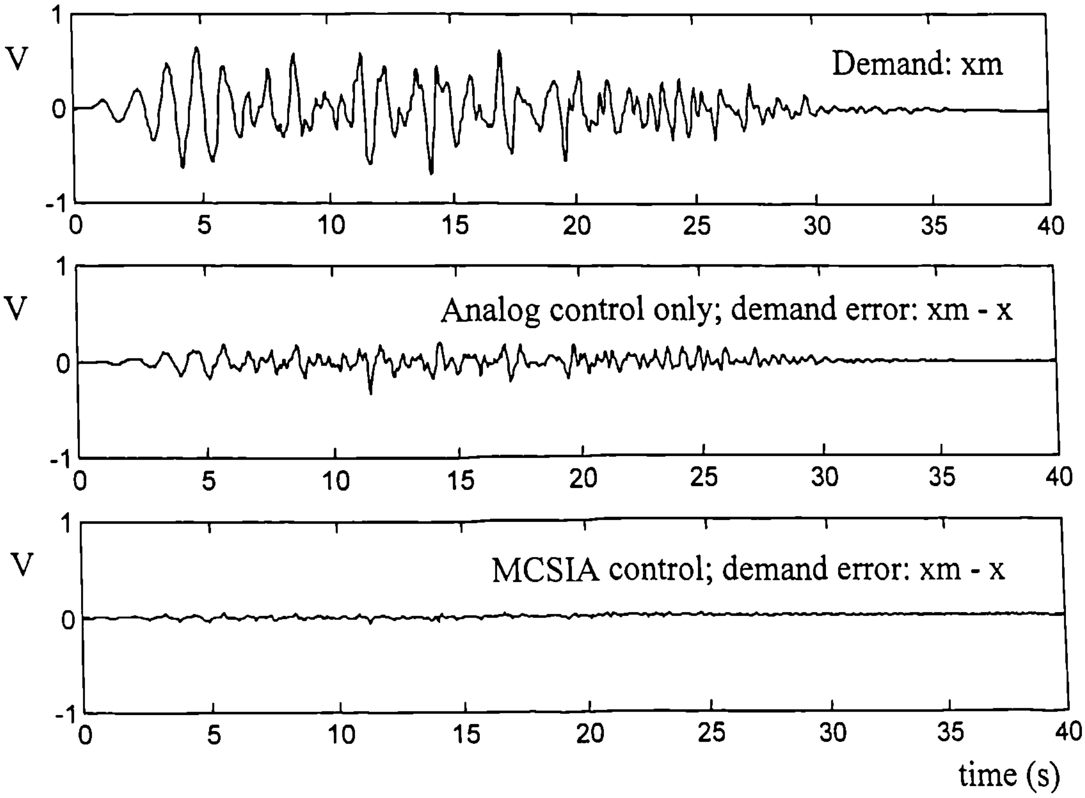


Fig. 6.16 Displacement errors for the single axis table at ISMES with and without MCS (from Stoten and Gomez, 1998)

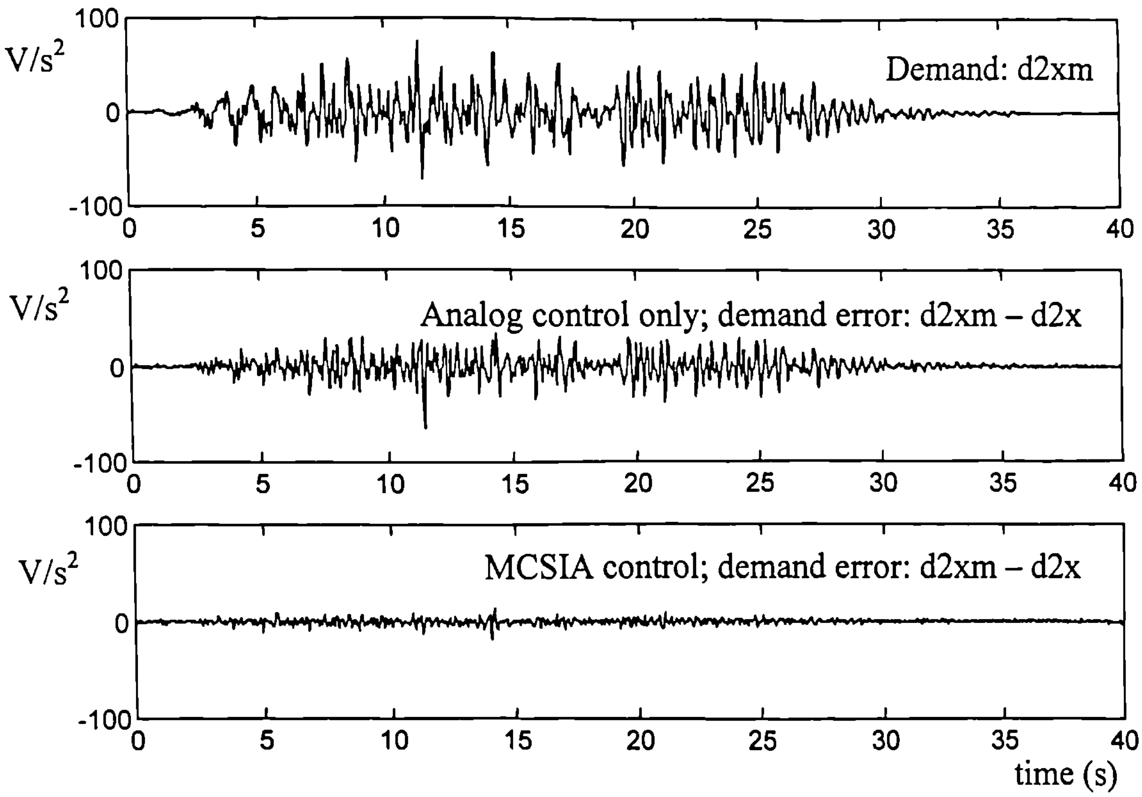


Fig. 6.17 Acceleration errors for the single axis table at ISMES with and without MCS (from Stoten and Gomez, 1998)

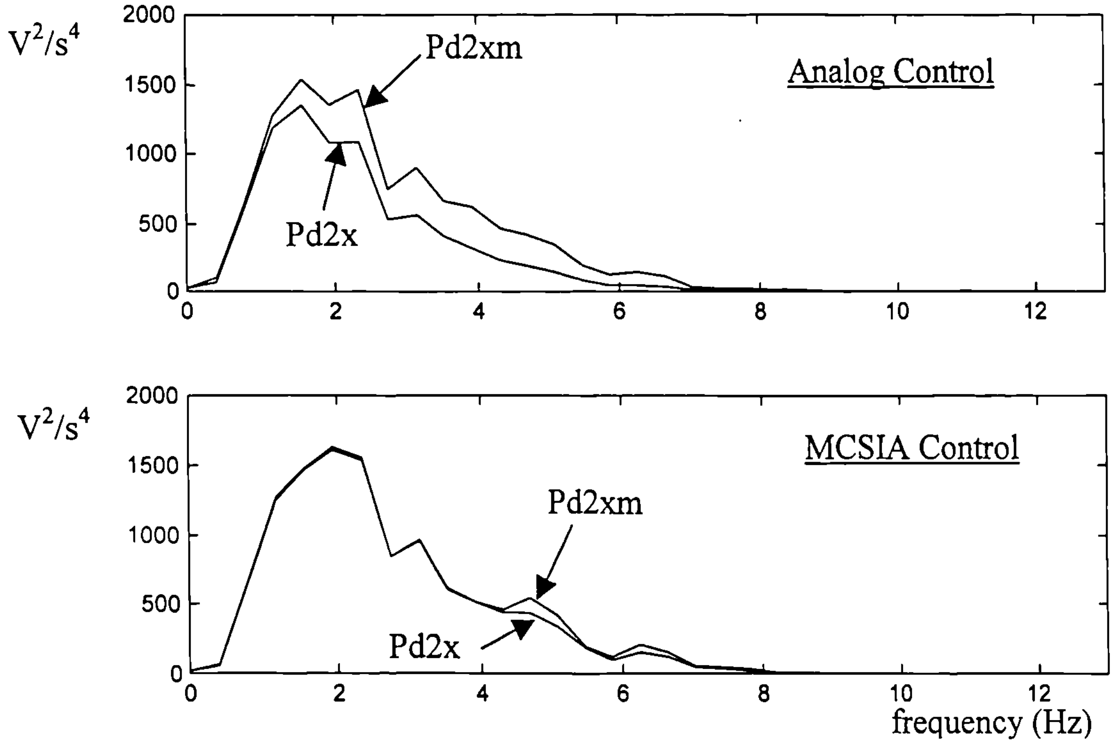


Fig. 6.18 Acceleration spectra of the single axis table at ISMES with and without MCS (from Stoten and Gomez, 1998)

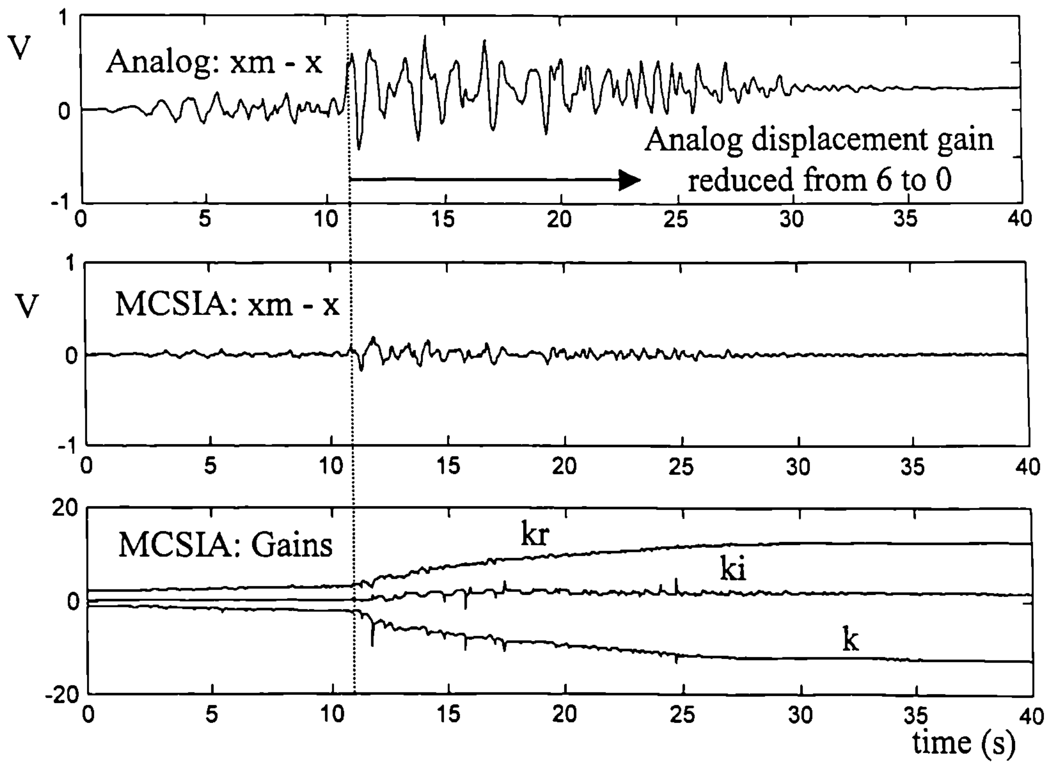


Fig. 6.19 Displacement errors of the single axis table at ISMES with and without MCS during a major parameter change (from Stoten and Gomez, 1998)

Chapter 7

Conclusions and Recommendations

7.1 Introduction

This thesis has described an extensive programme of work investigating the factors affecting earthquake shaking table experimentation, in particular comparing the performance of four different European shaking tables. For the first time several earthquake shaking tables have been compared to a common standard, and this research has produced much valuable information on the characteristics and performance of these shaking tables in particular, and of shaking tables in general. As a result of this work, recommendations have been made for making the best use of shaking tables for Earthquake Engineering research. Some of the possible future developments in shaking table control have also been outlined.

Shaking tables provide a test facility that will continue to be important in dynamic and seismic testing fields. The results of the comparison of the four shaking tables proved that all four tables could accurately reproduce various desired platform motions, so long as the shaking table test was carefully planned and was within the capacity of the table. This was possible even though the four tables had different capabilities and performances. Significant linear table-specimen interaction could also be largely controlled, provided that the operator was aware of the behaviour of the table and understood how the different methods of iterative earthquake matching software were best implemented. Nevertheless, shaking table-specimen interaction can still cause problems during shaking table tests, and should be carefully monitored whenever tests are performed.

This programme of characterisation tests has also given each of the shaking table groups concerned a greater understanding of their own shaking table, and the knowledge required to set up and control the table effectively when carrying out tests.

7.2 Key observations

During the work with the test specimen on the four shaking tables, and analysis of the results obtained, it became apparent that the operators of shaking tables need to be aware of a number of basic considerations. These are:

Test set-up

- Backlash in table bearings should be minimised by employing a regular maintenance schedule, and by carefully inspecting the table before each test is performed.
- The transducers in all actuators should be regularly calibrated in order to avoid actuator fighting.
- The shaking table control hardware should be tuned as accurately as possible before each test.
- Quality Assurance records should be kept of all test details.

Measurement and control of platform motion

- Before running a test on a shaking table, the cumulative power spectra of the required motions should be evaluated and the cut-off frequency of all the input and output signals should be set at the frequency below which 98% of the energy in the signal has been developed. This will avoid instability in the matching algorithms when trying to control the table to 50 Hz, for example, when the signal has little spectral content above 10 Hz.
- All platform motions (preferably both accelerations and displacements) should be measured during a test. If possible, this should include rotational platform motions as well as the translational ones.
- Rotational motions should be monitored so that it is possible to reconstruct the effective platform motion and the forces actually applied to the structure.
- All shaking table axes should be actively controlled by the matching software, even if the desired motion in any axis is zero.
- The matching software should use data from external accelerometers or displacement transducers to control the platform motion if it is important that the platform motion is reproduced as accurately as possible. Less stringent testing (for example undergraduate research) need only require the use of the transducers built into the table.

Specific points for matching

- Matching of test motions that do not comply with all of the displacement, velocity and acceleration limits of the shaking table should not be attempted. If the desired signal is beyond the capabilities of the table, then, the motion should be modified so that it can be achieved on the table. However, any modification of the signal should not affect the frequency components of the signal at the natural frequencies of the specimen being tested.

- The frequency range adopted during the processing of data as part of the matching process should be as wide as possible while still producing a numerically stable algorithm.

- If possible, the first iteration of any matching should include a precompensation stage (see page 6.5).

- If oil flow, or other repeatable non-linearity in the table, is expected to be a problem, then segmented time histories should be used in the matching process.

- At the extremes of performance any table is least likely to behave as a linear system, so the scaling of drive signals is least likely to be effective.

- Iterative non-linear matching techniques should be used when the linear matching algorithm shows signs of numerical instability.

- In the rare cases when the specimen to be tested will behave linearly throughout the test, the platform motion should be iteratively matched using non-linear matching techniques. However, the test should still be carefully monitored, as the table itself is not a linear system, and in any case the assumption that the specimen will behave linearly will need to be confirmed.

- When the specimen to be tested will become non-linear during the test, the specimen should be mounted on the platform, but restrained as much as possible so that it cannot move relative to the platform, and the time history should be matched at full-scale.

- If matching has to be performed without the specimen on the platform the specimen to be tested should be very small, from experience, less than about 10% of the mass of the platform. This will avoid subsequent problems with the platform motion once the specimen is mounted on the table.

- Even a relatively small specimen (about 10% of the mass of the platform) can cause significant table-specimen interaction, enough to seriously affect the response of the specimen.
- If the specimen on the platform has a mass greater than about 10% the mass of the platform, then the possible effects of table-specimen interaction on the platform motion and the test results should be considered.
- If significant specimen-table interaction is expected, enough additional instruments should be added to the platform to allow control of the interaction or at least provide a record of it.
- When testing highly non-linear models (bouncing/slipping blocks etc. that act as significant energy absorbers) it will become almost impossible to control the table behaviour without real-time control software such as MCS. Alternatively a massive table (about 10 to 20 times the mass of the specimen) should be used so that the specimen mass becomes relatively insignificant.
- Iterative spectral matching should be used to create the platform motions used in seismic qualification testing where required response spectra are specified.
- Iterative displacement matching produces the best overall results for both platform displacements and accelerations. However, if very accurate reproduction of platform accelerations are required iterative acceleration matching should be used.

Analysis of results

- When comparing test results with the theoretical response of the specimen to the platform motion, the actual platform motion should be used rather than the desired platform motion.

7.3 Conclusions

The major contribution made by this research is that after examining in depth the performance of four shaking tables it has been possible to characterise their performance in the time and frequency domains and identify the problems associated with shaking table experimentation. It is now possible to control the errors in platform motion more efficiently, or at least take then into account in the analysis of any shaking table test results.

This research had seven key aims (see §1.2) and these have all been generally satisfied, although in some instances not completely achieved.

The results of the research programme can be summarised as follows:

1. A detailed characterisation of the dynamic performance of several shaking tables, both with and without a specimen attached, has been produced in a consistent manner in both time and frequency domains.
2. A systematic methodology for regularly assessing the performance of an earthquake shaking table has been developed (§5.5.6).
3. The strengths and weaknesses of the four tables have been studied, and in many cases improvements to the hardware and software control systems have been made to improve the overall performance of the tables (§5.6.2, §5.6.3 and §5.6.4).
4. The ability of the tables to respond accurately to input signals has been studied in detail. The efficiency of the control software and the mechanical control systems in avoiding unwanted platform motions has been examined, and some new methodologies for controlling the undesired motions have been developed (Chapter 6).
5. The problem of shaking table-specimen interaction has been investigated to a limited extent for linear specimens. Under certain circumstances this interaction has been found to have a significant effect on the behaviour of a specimen.
6. The iterative software used by the four laboratories for shaking table testing has been compared, and new iterative matching software, based on the best systems found, is being developed for use in the four laboratories.
7. The testing techniques that worked most efficiently at the four laboratories were compared, and many of the potential pitfalls in shaking table testing have been identified for researchers who wish to perform shaking table tests in the future (§7.2).

7.4 Recommendations for further study

These investigations have clarified the need for further research on some outstanding problems. In particular, there is a need to develop a significantly non-linear specimen to test the ability of any iterative matching techniques to cope with major changes in specimen-table interaction during a test. Further development tests of any real-time control system also need to be performed with a significantly non-linear specimen on the platform so that such a system can be tested up to its performance limits. The initial tests of a specific real-time control technique (MCS) for shaking tables have been very encouraging, but significant additional research still has to be performed before such a technique can be used exclusively to control shaking tables.

Although real-time control techniques are likely to be used to control shaking table in the future there is currently still a need to enhance and extend the existing iterative matching software that is used by the four shaking table laboratories. The use of non-square transfer function matrices within the matching process has been discussed in Chapter 6 but other techniques also need to be investigated further. The natural frequency of the Bristol shaking table platform and its horizontal actuators is around 16 Hz mainly due to the oil column resonance effects, and this is reflected in measured platform response spectra. The control algorithms had insufficient energy to utilise around this frequency, and were therefore unable to eliminate the effects of this resonance. Further work is needed, using higher frequency bandwidth input signals and lower natural frequency specimens, to explore this phenomenon. In particular, it would be of interest to study the effects of injecting into the drive signal a small amount of random noise around the shaking table natural frequency to act as a seed for the control algorithms.

In addition to the continuing development and validation of shaking table control methodologies it would be extremely useful to develop is a set of standard time histories for system performance evaluation. These time histories should cater for the wide range of possible performance characteristics of a shaking table, and would thereby provide a common benchmark against which almost any shaking table could be assessed. The unscaled El Centro and Kalamata records used for this research are not ideally suited for the role of standard time histories; however, by scaling these two records it should be possible to test any shaking table up to its performance limits.

Most of the testing carried out on the Bristol shaking table, at the moment, is on scale models where the applied shakes are scaled with the frequency content shifted towards the higher frequencies. Frequency-scaled versions of the El Centro and Kalamata excitations should therefore also be used for testing the table performance under its normal operating conditions. A new series of time histories should be defined that includes these higher frequencies, and the tests that were performed during this research project ideally need to be repeated with the new time histories. The time-history matching studies of the Athens and ISMES tables also need to be extended to cover higher frequency motions.

During this research there was insufficient time to study the performance of any of the shaking tables with non-linear specimens. This remains a pertinent topic for further study. There is also a need for further studies on table-specimen interaction, particularly considering the performance of shaking tables with non-linear specimens.

In conclusion, the results of the comparison of the shaking tables proved that all four shaking tables could accurately reproduce various desired platform motions and that significant linear table-specimen interaction could also be controlled. This was possible even though the four tables had markedly different capabilities and performances. Shaking tables are excellent research tools for studying the effects of earthquake motion on all sorts of structures, although like all tools they should be used safely and correctly. This research basically developed from a need to develop more efficient ways for performing shaking table testing, and as these techniques are developed further we are likely to see shaking tables being used ever more extensively in the future.

Appendix A

Tests performed at the four laboratories

A.1 Bristol laboratory

The full series of frequency response test results for the Bristol table have already been published (Crewe & Taylor, 1994), but a summary of the tests performed is included here (table A.1). The impulse and random signal test numbers are listed, along with details of the type of input, the test type (including whether externally mounted or the internal accelerometers in the table were used for the measurements), and the loading on the table.

Table A.1 List of Impulse and Random signal tests performed at Bristol (Nov. 1993)

Test No.	Axis being driven and signal type being used	Axis being recorded and which instruments being used	Table loading
1	X axis Impulse input	X axis table motion (external accels)	Bare Table
2	X axis Impulse input	Y axis table motion (external accels)	Bare Table
3	X axis Impulse input	Z axis table motion (external accels)	Bare Table
4	X axis Impulse input	P axis table motion (internal accels)	Bare Table
5	X axis Impulse input	R axis table motion (internal accels)	Bare Table
6	X axis Impulse input	W axis table motion (internal accels)	Bare Table
7	Y axis Impulse input	X axis table motion (internal accels)	Bare Table
8	Y axis Impulse input	X axis table motion (external accels)	Bare Table
9	Y axis Impulse input	Y axis table motion (external accels)	Bare Table
10	Y axis Impulse input	Z axis table motion (external accels)	Bare Table
11	Y axis Impulse input	P axis table motion (internal accels)	Bare Table
12	Y axis Impulse input	R axis table motion (internal accels)	Bare Table
13	Y axis Impulse input	W axis table motion (internal accels)	Bare Table
14	Z axis Impulse input	X axis table motion (external accels)	Bare Table
15	Z axis Impulse input	Y axis table motion (external accels)	Bare Table
16	Z axis Impulse input	Z axis table motion (external accels)	Bare Table
17	Z axis Impulse input	P axis table motion (internal accels)	Bare Table
18	Z axis Impulse input	R axis table motion (internal accels)	Bare Table
19	Z axis Impulse input	W axis table motion (internal accels)	Bare Table
20	P axis Impulse input	P axis table motion (internal accels)	Bare Table
21	R axis Impulse input	R axis table motion (internal accels)	Bare Table
22	W axis Impulse input	W axis table motion (internal accels)	Bare Table
23	X axis Impulse input	X axis table motion (external accels)	Bare Table
24	Y axis Impulse input	Y axis table motion (external accels)	Bare Table
25	Z axis Impulse input	Z axis table motion (external accels)	Bare Table
26	X axis Impulse input	X axis table motion (external accels)	4 tonnes
27	Y axis Impulse input	Y axis table motion (external accels)	4 tonnes
28	Z axis Impulse input	Z axis table motion (external accels)	4 tonnes

Test No.	Axis being driven and signal type being used	Axis being recorded and which instruments being used	Table loading
29	P axis Impulse input	P axis table motion (internal accels)	4 tonnes
30	R axis Impulse input	R axis table motion (internal accels)	4 tonnes
31	W axis Impulse input	W axis table motion (internal accels)	4 tonnes
32	X axis Random input	X axis table motion (external accels)	4 tonnes
33	X axis Random input	X axis table motion (external accels)	4 tonnes
34	Y axis Random input	Y axis table motion (external accels)	4 tonnes
35	Z axis Random input	Z axis table motion (external accels)	4 tonnes
36	X axis Impulse input	X axis table motion (external accels)	8 tonnes
37	Y axis Impulse input	Y axis table motion (external accels)	8 tonnes
38	Z axis Impulse input	Z axis table motion (external accels)	8 tonnes
39	P axis Impulse input	P axis table motion (internal accels)	8 tonnes
40	R axis Impulse input	R axis table motion (internal accels)	8 tonnes
41	W axis Impulse input	W axis table motion (internal accels)	8 tonnes
42	X axis Random input	X axis table motion (external accels)	8 tonnes
43	Y axis Random input	Y axis table motion (external accels)	8 tonnes
44	Z axis Random input	Z axis table motion (external accels)	8 tonnes
45	Y axis table motion	Y axis frame motion (external Accels)	5 tonne frame
46	Y axis Random input	Y axis table motion (external Accels)	5 tonne frame
47	X axis table motion	X axis frame motion (external Accels)	5 tonne frame
48	X axis Random input	X axis table motion (external Accels)	5 tonne frame

The full series of time history test results for the Bristol table have already been published (Crewe & Taylor, 1994), but a summary of the tests performed is included here (table A.2). The seismic signal being matched is listed, along with details of the loading on the table, the number of iterations performed, the type of signal used at the start of the matching, and whether the acceleration or displacement time history was being matched, along with any comments as to how the test proceeded.

Table A.2 List of time history tests performed at Bristol (Nov. 1993)

Signal	Table Loading	Iteration	Initial signal used	Type of match	Comments
EL Centro	Bare table				
		Iteration 0	Accel Start	Accel Match	
		Iteration 1	Accel Start	Accel Match	
EL Centro	Bare table				
		Iteration 0	Accel Start	Accel Match	
		Iteration 1	Accel Start	Accel Match	
		Iteration 1a	Accel Start	Disp Match	
		Iteration 2	Accel Start	Accel Match	
		Iteration 3	Accel Start	Accel Match	
		Iteration 4	Accel Start	Accel Match	
		Iteration 5	Accel Start	Accel Match	
		Iteration 6	Accel Start	Accel Match	

Signal	Table Loading	Iteration	Initial signal used	Type of match	Comments
EL Centro	Bare table				
		Iteration 0	Disp Start	Disp Match	Truncated test
		Iteration 1	Disp Start	Disp Match	Truncated test
EL Centro	Bare table				
		Iteration 0	Disp Start	Accel Match	
		Iteration 1	Disp Start	Accel Match	
EL Centro	Bare table				
		Iteration 0	Disp Start	Disp Match	
		Iteration 1	Disp Start	Disp Match	
		Iteration 2	Disp Start	Disp Match	
EL Centro	Bare table				
		Iteration 0	Disp Start	Accel Match	
		Iteration 1	Disp Start	Accel Match	
		Iteration 2	Disp Start	Accel Match	
		Iteration 3	Disp Start	Accel Match	
		Iteration 4	Disp Start	Accel Match	
		Iteration 5	Disp Start	Accel Match	
EL Centro	4 tonnes				
		Iteration 0	Disp Start	Accel Match	
		Iteration 1	Disp Start	Accel Match	
		Iteration 2	Disp Start	Accel Match	
		Iteration 2a	Disp Start	Accel Match	
		Iteration 2b	Disp Start	Accel Match	
EL Centro	8 tonnes				
		Iteration 0	Disp Start	Accel Match	
		Iteration 1	Disp Start	Accel Match	
		Iteration 1a	Disp Start	Accel Match	
EL Centro	5 tonne frame				
		Iteration 0	Disp Start	Accel Match	
		Iteration 1	Disp Start	Accel Match	
Kalamata	Bare table				
		Iteration 0	Accel Start	Accel Match	
		Iteration 1	Accel Start	Accel Match	
		Iteration 2	Accel Start	Accel Match	
		Iteration 3	Accel Start	Accel Match	
Kalamata	Bare table				
		Iteration 0	Disp Start	Disp Match	
		Iteration 1	Disp Start	Disp Match	
		Iteration 2	Disp Start	Disp Match	
		Iteration 3	Disp Start	Disp Match	
		Iteration 4	Disp Start	Disp Match	
		Iteration 5	Disp Start	Disp Match	
		Iteration 6	Disp Start	Disp Match	
Kalamata	Bare table				
		Iteration 0	Vel Start	Accel Match	
		Iteration 1	Vel Start	Accel Match	
		Iteration 2	Vel Start	Accel Match	
		Iteration 3	Vel Start	Accel Match	
		Iteration 4	Vel Start	Accel Match	
		Iteration 6	Vel Start	Accel Match	Non linear Match
		Iteration 7	Vel Start	Accel Match	Non linear Match
Kalamata	4 tonnes				

Signal	Table Loading	Iteration	Initial signal used	Type of match	Comments
		Iteration 2a	Disp Start	Accel Match	
Kalamata	8 tonnes				
		Iteration 0	Disp Start	Accel Match	
		Iteration 1	Disp Start	Accel Match	
		Iteration 2	Disp Start	Accel Match	
		Iteration 2a	Disp Start	Accel Match	
Kalamata	8 tonnes				
		Iteration 0	Disp Start	Disp Match	
		Iteration 1	Disp Start	Disp Match	
		Iteration 2	Disp Start	Disp Match	
		Iteration 2a	Disp Start	Accel Match	
Kalamata	8 tonnes				
		Iteration 0	Disp Start	Disp Match	
		Iteration 1	Disp Start	Disp Match	
		Iteration 2a	Disp Start	Accel Match	
Kalamata	5 tonne frame				
		Iteration 0	Disp Start	Accel Match	
		Iteration 1	Disp Start	Accel Match	
		Iteration 2a	Disp Start	Accel Match	

A.2 Athens laboratory

The full series of frequency response test results for the Athens table have already been published (Crewe & Taylor, 1994), but a summary of the tests performed is included here (table A.3). Measurements were taken to explore the frequency response of the Athens shaking table when carrying a variety of rigid and flexible payloads. The impulse and random signal test numbers are listed, along with details of the type of input, the condition of tuning of the table, the type of feedback control being used in the hardware control system, the signal being recorded, and the loading on the table.

Table A.3 List of Impulse and Random signal tests performed at Athens (Dec. 1993)

Test No.	Axis being driven and signal type being used	Condition of tuning	Mode of control	Axis being recorded	Table loading
1	Y axis random input	System tuned	Acceleration	Y axis table motion	Bare Table
2	Y axis impulse input	No further tuning	Acceleration	Y axis table motion	Bare Table
3	X axis random input	No further tuning	Acceleration	X axis table motion	Bare Table
4	X axis impulse input	No further tuning	Acceleration	X axis table motion	Bare Table
5	Z axis random input	No further tuning	Acceleration	Z axis table motion	Bare Table
6	Z axis impulse input	No further tuning	Acceleration	Z axis table motion	Bare Table
7	Z axis random input	No further tuning	Acceleration + external input	Z axis table motion	Bare Table
8	Z axis impulse input	No further tuning	Acceleration + external input	Z axis table motion	Bare Table
9	Z axis random input	No further tuning	Acceleration + external input	Z axis table motion	Bare Table
10	R axis random input	No further tuning	Acceleration	R axis table motion	Bare Table
11	R axis impulse input	No further tuning	Acceleration	R axis table motion	Bare Table
12	P axis random input	No further tuning	Acceleration	P axis table motion	Bare Table
13	P axis impulse input	No further tuning	Acceleration	P axis table motion	Bare Table
14	W axis random input	No further tuning	Acceleration	W axis table motion	Bare Table
15	W axis impulse input	No further tuning	Acceleration	W axis table motion	Bare Table
16	Y axis random input	No further tuning	Acceleration	R axis table motion	Bare Table
17	Y axis random input	No further tuning	Acceleration	P axis table motion	Bare Table
18	Y axis random input	No further tuning	Acceleration	W axis table motion	Bare Table
19	Y axis random input	No further tuning	Acceleration	X axis table motion	Bare Table
20	Y axis random input	No further tuning	Acceleration	Z axis table motion	Bare Table
21	X axis random input	No further tuning	Acceleration	R axis table motion	Bare Table
22	X axis random input	No further tuning	Acceleration	P axis table motion	Bare Table
23	X axis random input	No further tuning	Acceleration	W axis table motion	Bare Table
24	X axis random input	No further tuning	Acceleration	Y axis table motion	Bare Table
25	X axis random input	No further tuning	Acceleration	Z axis table motion	Bare Table

Test No.	Axis being driven and signal type being used	Condition of tuning	Mode of control	Axis being recorded	Table loading
26	Z axis random input	No further tuning	Acceleration	R axis table motion	Bare Table
27	Z axis random input	No further tuning	Acceleration	P axis table motion	Bare Table
28	Z axis random input	No further tuning	Acceleration	W axis table motion	Bare Table
29	Z axis random input	No further tuning	Acceleration	Y axis table motion	Bare Table
30	Z axis random input	No further tuning	Acceleration	X axis table motion	Bare Table
31	Y axis random input	No further tuning	Displacement	Y axis table motion	Bare Table
32	Y axis random input	No further tuning	Direct Displacement	Y axis table motion	Bare Table
33	Y axis random input	No further tuning	Acceleration + external input	Y axis table motion	Bare Table
34	Y axis impulse input	No further tuning	Acceleration + external input	Y axis table motion	Bare Table
35	R axis random input	No further tuning	Acceleration	Y axis table motion	Bare Table
36	R axis random input	No further tuning	Acceleration	Z axis table motion	Bare Table
37	P axis random input	No further tuning	Acceleration	X axis table motion	Bare Table
38	P axis random input	No further tuning	Acceleration	Z axis table motion	Bare Table
39	W axis random input	No further tuning	Acceleration	X axis table motion	Bare Table
40	W axis random input	No further tuning	Acceleration	Y axis table motion	Bare Table
41	Z axis random input	No further tuning	Acceleration	Z axis table motion	4 tonnes
42	P axis random input	No further tuning	Acceleration	P axis table motion	4 tonnes
43	R axis random input	No further tuning	Acceleration	R axis table motion	4 tonnes
44	X axis random input	No further tuning	Acceleration	X axis table motion	4 tonnes
45	W axis random input	No further tuning	Acceleration	W axis table motion	4 tonnes
46	Y axis random input	No further tuning	Acceleration	Y axis table motion	4 tonnes
47	Z axis random input	No further tuning	Acceleration	Z axis table motion	8 tonnes
48	P axis random input	No further tuning	Acceleration	P axis table motion	8 tonnes
49	R axis random input	No further tuning	Acceleration	R axis table motion	8 tonnes
50	X axis random input	No further tuning	Acceleration	X axis table motion	8 tonnes
51	W axis random input	No further tuning	Acceleration	W axis table motion	8 tonnes
52	Y axis random input	No further tuning	Acceleration	Y axis table motion	8 tonnes
53	Y axis random input	System re-tuned	Acceleration	Y axis table motion	8 tonnes

Test No.	Axis being driven and signal type being used	Condition of tuning	Mode of control	Axis being recorded	Table loading
54	R axis random input	No further tuning	Acceleration	R axis table motion	8 tonnes
55	Y axis random input	No further tuning	Acceleration	Y axis table motion	8 tonnes
56	Z axis random input	System re-tuned	Acceleration	Z axis table motion	Bare Table
57	Y & Z axis random input	No further tuning	Acceleration	Z axis table motion	Bare Table
58	X, Y &Z axis random input	No further tuning	Acceleration	Z axis table motion	Bare Table
59	Z axis random input	No further tuning	Acceleration	Z axis table motion	8 tonnes
60	Z axis random input	System re-tuned	Acceleration	Z axis table motion	8 tonnes
61	Z axis random input	No further tuning	Acceleration + static lift	Z axis table motion	8 tonnes
62	Y axis random input	No further tuning	Acceleration	Y axis table motion	5 tonne frame
62a	Y axis random input	System re-tuned	Acceleration	Y axis table motion	5 tonne frame
62b	Y axis random input	System re-tuned	Acceleration	Y axis table motion	5 tonne frame
63	X axis random input	No further tuning	Acceleration	X axis table motion	5 tonne frame
64	X axis random input	No further tuning	Acceleration	X axis table motion	5 tonne frame
65	Z axis random input	No further tuning	Acceleration	Z axis table motion	5 tonne frame
66	R axis random input	No further tuning	Acceleration	R axis table motion	5 tonne frame
67	Y axis random input	No further tuning	Acceleration	R axis table motion	5 tonne frame
68	W axis random input	No further tuning	Acceleration	W axis table motion	5 tonne frame
69	Y axis random input	No further tuning	Acceleration	W axis table motion	5 tonne frame
70	X axis random input	No further tuning	Acceleration	W axis table motion	5 tonne frame
71	X axis random input	No further tuning	Acceleration	R axis table motion	5 tonne frame
72	Y axis random input	No further tuning	Acceleration	Y axis specimen	5 tonne frame
73	X axis random input	No further tuning	Acceleration	X axis specimen	5 tonne frame
74	Y axis random input	No further tuning	Acceleration	Y axis specimen	7 tonne frame
75	Y axis random input	No further tuning	Acceleration	Y axis specimen	10 tonne frame
76	Y axis random input	No further tuning	Acceleration	Y axis specimen	10 tonne frame
77	Y axis random input	No further tuning	Acceleration	X axis specimen	10 tonne frame
78	Y axis random input	No further tuning	Acceleration	X axis specimen	10 tonne frame
79	X axis random input	No further tuning	Acceleration	X axis table motion	10 tonne frame

Test No.	Axis being driven and signal type being used	Condition of tuning	Mode of control	Axis being recorded	Table loading
80	X axis random input	No further tuning	Acceleration	X axis specimen	10 tonne frame
81	Y axis random input	No further tuning	Displacement	Y axis table motion	10 tonne frame
82	X axis random input	No further tuning	Displacement	X axis table motion	10 tonne frame
83	Z axis random input	No further tuning	Displacement	Z axis table motion	10 tonne frame

The full series of time history test results for the Athens table have already been published (Crewe & Taylor, 1994), but a summary of the tests performed is included here (table A.4). The seismic signal being matched is listed, along with details of the loading on the table, the number of iterations performed, the type of signal used at the start of the matching, and whether the acceleration or displacement time history was being matched.

Table A.4 List of time history tests performed at Athens (Dec. 1993)

Signal	Table Loading	Iteration	Initial signal used	Type of match
EL Centro	Bare table			
		Iteration 0	Accel Start	Accel Match
		Iteration 1	Accel Start	Accel Match
Kalamata	Bare table			
		Iteration 0	Accel Start	Accel Match
		Iteration 1	Accel Start	Accel Match
Kalamata	4 tonnes			
		Iteration 0	Accel Start	Accel Match
		Iteration 1	Accel Start	Accel Match
		Iteration 2	Accel Start	Accel Match
EL Centro	8 tonnes			
		Iteration 0	Accel Start	Accel Match
		Iteration 1	Accel Start	Accel Match
Kalamata	8 tonnes			
		Iteration 0	Accel Start	Accel Match
		Iteration 1	Accel Start	Accel Match
Kalamata	10 tonne frame			
		Iteration 0	Accel Start	Accel Match
		Iteration 1	Accel Start	Accel Match

A.3 ISMES laboratory

The full series of frequency response test results for the ISMES table have already been published (ISMES, 1996), but a summary of the tests performed is included here (table A.5). Measurements were taken to explore the frequency response of the ISMES shaking table when carrying a variety of rigid and flexible payloads. The signal test numbers are listed, along with details of the axis excited and the signal being used, the signal being recorded, the condition of tuning of the table, and the loading on the table.

Table A.5 List of Impulse and Random signal tests performed at ISMES (Nov. 1995)

Test No.	Axis being driven	Axis being recorded	Condition of tuning	Table loading
1	X axis random input	X axis table motion	System tuned	Bare Table
2	Y axis random input	Y axis table motion	No further tuning	Bare Table
3	Z axis random input	Z axis table motion	No further tuning	Bare Table
4	W axis random input	W axis table motion	No further tuning	Bare Table
5	P axis random input	P axis table motion	No further tuning	Bare Table
6	R axis random input	R axis table motion	No further tuning	Bare Table
7	X axis random input	X axis table motion	No further tuning	8 tonnes
8	Y axis random input	Y axis table motion	No further tuning	8 tonnes
9	Z axis random input	Z axis table motion	No further tuning	8 tonnes
10	W axis random input	W axis table motion	No further tuning	8 tonnes
11	P axis random input	P axis table motion	No further tuning	8 tonnes
12	R axis random input	R axis table motion	No further tuning	8 tonnes
13	X axis random input	X axis table motion	No further tuning	8 tonne frame
14	Y axis random input	Y axis table motion	No further tuning	8 tonne frame
15	Z axis random input	Z axis table motion	No further tuning	8 tonne frame
16	W axis random input	W axis table motion	No further tuning	8 tonne frame
17	P axis random input	P axis table motion	No further tuning	8 tonne frame
18	R axis random input	R axis table motion	No further tuning	8 tonne frame
19	X axis random input	X axis table motion	System tuned	8 tonne frame
20	X axis random input	X axis table motion	System tuned	8 tonne frame
21	X axis random input	X axis table motion	No further tuning	8 tonne frame
22	X axis random input	Y axis table motion	No further tuning	8 tonne frame
23	X axis random input	Z axis table motion	No further tuning	8 tonne frame
24	X axis random input	W axis table motion	No further tuning	8 tonne frame
25	X axis random input	P axis table motion	No further tuning	8 tonne frame
26	X axis random input	R axis table motion	No further tuning	8 tonne frame
27	Y axis random input	X axis table motion	No further tuning	8 tonne frame
28	Y axis random input	Y axis table motion	No further tuning	8 tonne frame
29	Y axis random input	Z axis table motion	No further tuning	8 tonne frame
30	Y axis random input	W axis table motion	No further tuning	8 tonne frame
31	Y axis random input	P axis table motion	No further tuning	8 tonne frame
32	Y axis random input	R axis table motion	No further tuning	8 tonne frame
33	Z axis random input	X axis table motion	No further tuning	8 tonne frame
34	Z axis random input	Y axis table motion	No further tuning	8 tonne frame
35	Z axis random input	Z axis table motion	No further tuning	8 tonne frame
36	Z axis random input	W axis table motion	No further tuning	8 tonne frame
37	Z axis random input	P axis table motion	No further tuning	8 tonne frame
38	Z axis random input	R axis table motion	No further tuning	8 tonne frame
39	W axis random input	X axis table motion	No further tuning	8 tonne frame
40	W axis random input	Y axis table motion	No further tuning	8 tonne frame
41	W axis random input	Z axis table motion	No further tuning	8 tonne frame
42	W axis random input	W axis table motion	No further tuning	8 tonne frame
43	W axis random input	P axis table motion	No further tuning	8 tonne frame
44	W axis random input	R axis table motion	No further tuning	8 tonne frame
45	P axis random input	X axis table motion	No further tuning	8 tonne frame
46	P axis random input	Y axis table motion	No further tuning	8 tonne frame
47	P axis random input	Z axis table motion	No further tuning	8 tonne frame
48	P axis random input	W axis table motion	No further tuning	8 tonne frame
49	P axis random input	P axis table motion	No further tuning	8 tonne frame
50	P axis random input	R axis table motion	No further tuning	8 tonne frame
51	R axis random input	X axis table motion	No further tuning	8 tonne frame

Test No.	Axis being driven	Axis being recorded	Condition of tuning	Table loading
52	R axis random input	Y axis table motion	No further tuning	8 tonne frame
53	R axis random input	Z axis table motion	No further tuning	8 tonne frame
54	R axis random input	W axis table motion	No further tuning	8 tonne frame
55	R axis random input	P axis table motion	No further tuning	8 tonne frame
56	R axis random input	R axis table motion	No further tuning	8 tonne frame
57	Z axis random input ($0.01 m/s^2$)	Z axis table motion	No further tuning	Bare Table
58	Z axis random input ($0.03 m/s^2$)	Z axis table motion	No further tuning	Bare Table
59	Z axis random input ($0.04 m/s^2$)	Z axis table motion	No further tuning	Bare Table
60	X axis sine sweep input	X axis table motion	No further tuning	8 tonne frame
61	X axis sine sweep input	Y axis table motion	No further tuning	8 tonne frame
62	X axis sine sweep input	Z axis table motion	No further tuning	8 tonne frame
63	X axis sine sweep input	X specimen motion	No further tuning	8 tonne frame
64	X axis sine sweep input	Y specimen motion	No further tuning	8 tonne frame
65	X axis sine sweep input	Z specimen motion	No further tuning	8 tonne frame
66	X axis sine sweep input	X specimen motion	No further tuning	8 tonne frame
67	X axis sine sweep input	Z specimen motion	No further tuning	8 tonne frame
68	X axis sine sweep input	Y specimen motion	No further tuning	8 tonne frame
69	X axis sine sweep input	Z specimen motion	No further tuning	8 tonne frame
70	Y axis sine sweep input	X axis table motion	No further tuning	8 tonne frame
71	Y axis sine sweep input	Y axis table motion	No further tuning	8 tonne frame
72	Y axis sine sweep input	Z axis table motion	No further tuning	8 tonne frame
73	Y axis sine sweep input	X specimen motion	No further tuning	8 tonne frame
74	Y axis sine sweep input	Y specimen motion	No further tuning	8 tonne frame
75	Y axis sine sweep input	Z specimen motion	No further tuning	8 tonne frame
76	Y axis sine sweep input	X specimen motion	No further tuning	8 tonne frame
77	Y axis sine sweep input	Z specimen motion	No further tuning	8 tonne frame
78	Y axis sine sweep input	Y specimen motion	No further tuning	8 tonne frame
79	Y axis sine sweep input	Z specimen motion	No further tuning	8 tonne frame

The full series of time history test results for the ISMES table have already been published (ISMES, 1996), but a summary of the tests performed is included here (table A.6). The test name is listed, along with details of the loading on the table, the number of axes being actively controlled, whether the acceleration or displacement time history was being matched, and the frequency of the filter used on the input signal to the table.

Table A.6 Time history tests performed at ISMES (Nov. 1995)

Test	Table Condition	Number of Axes Controlled	Control Mode	Drive Filter (Hz)
Kalamata	Bare	3	Acceleration	0.5 - 35
Kala8t0	Rigid Payload	0	Acceleration	0.5 - 35
Kala8t3	Rigid Payload	3	Acceleration	0.5 - 35
Kala8t6	Rigid Payload	6	Acceleration	0.3 - 35
Kala8t6d	Rigid Payload	6	Displacement	0.1 - 20
Flex3	Flexible Payload	3	Acceleration	0.2 - 35
Flex6d	Flexible Payload	6	Displacement	0.1 - 20
Flex6	Flexible Payload	6	Acceleration	0.2 - 35

A.4 LNEC laboratory

The full series of frequency response test results for the LNEC table have already been published (LNEC, 1996), but a summary of the tests performed is included here (table A.7). Measurements were taken to explore the frequency response of the bare LNEC shaking table and the table when carrying a flexible payload. The signal test numbers are listed, along with details of the axis excited and signal being used, the signal being recorded, the condition of tuning of the table, and the loading on the table.

Table A.7 List of Impulse and Random signal tests performed at LNEC (Feb. 1996)

Test No.	Axis being driven	Axis being recorded	Condition of tuning	Table loading
1	Y axis random input	Y axis table motion	System not tuned	Bare Table
2	Y axis random input	Z axis table motion	No further tuning	Bare Table
3	Y axis random input	X axis table motion	No further tuning	Bare Table
4	Y axis random input	P axis table motion	No further tuning	Bare Table
5	Y axis random input	R axis table motion	No further tuning	Bare Table
6	Y axis random input	W axis table motion	No further tuning	Bare Table
7	Z axis random input	Z axis table motion	No further tuning	Bare Table
8	Z axis random input	Y axis table motion	No further tuning	Bare Table
9	Z axis random input	X axis table motion	No further tuning	Bare Table
10	Z axis random input	P axis table motion	No further tuning	Bare Table
11	Z axis random input	R axis table motion	No further tuning	Bare Table
12	Z axis random input	W axis table motion	No further tuning	Bare Table
13	X axis random input	X axis table motion	No further tuning	Bare Table
14	X axis random input	Y axis table motion	No further tuning	Bare Table
15	X axis random input	Z axis table motion	No further tuning	Bare Table
16	X axis random input	P axis table motion	No further tuning	Bare Table
17	X axis random input	R axis table motion	No further tuning	Bare Table
18	X axis random input	W axis table motion	No further tuning	Bare Table
19	Y axis random input	Y axis table disps.	No further tuning	Bare Table

Test No.	Axis being driven	Axis being recorded	Condition of tuning	Table loading
20	Z axis random input	Z axis table disps.	No further tuning	Bare Table
21	X axis random input	X axis table disps.	No further tuning	Bare Table
22	Y axis random input	Y axis table motion	System tuned	Bare Table
23	Y axis random input	Z axis table motion	No further tuning	Bare Table
24	Y axis random input	X axis table motion	No further tuning	Bare Table
25	Y axis random input	P axis table motion	No further tuning	Bare Table
26	Y axis random input	R axis table motion	No further tuning	Bare Table
27	Y axis random input	W axis table motion	No further tuning	Bare Table
28	Z axis random input	Z axis table motion	No further tuning	Bare Table
29	Z axis random input	Y axis table motion	No further tuning	Bare Table
30	Z axis random input	X axis table motion	No further tuning	Bare Table
31	Z axis random input	P axis table motion	No further tuning	Bare Table
32	Z axis random input	R axis table motion	No further tuning	Bare Table
33	Z axis random input	W axis table motion	No further tuning	Bare Table
34	X axis random input	X axis table motion	No further tuning	Bare Table
35	X axis random input	Y axis table motion	No further tuning	Bare Table
36	X axis random input	Z axis table motion	No further tuning	Bare Table
37	X axis random input	P axis table motion	No further tuning	Bare Table
38	X axis random input	R axis table motion	No further tuning	Bare Table
39	X axis random input	W axis table motion	No further tuning	Bare Table
40	Y axis random input	Y axis table motion	System tuned	8 tonne frame
41	Y axis random input	Z axis table motion	No further tuning	8 tonne frame
42	Y axis random input	X axis table motion	No further tuning	8 tonne frame
43	Y axis random input	P axis table motion	No further tuning	8 tonne frame
44	Y axis random input	R axis table motion	No further tuning	8 tonne frame
45	Y axis random input	W axis table motion	No further tuning	8 tonne frame
46	Z axis random input	Z axis table motion	No further tuning	8 tonne frame
47	Z axis random input	Y axis table motion	No further tuning	8 tonne frame
48	Z axis random input	X axis table motion	No further tuning	8 tonne frame
49	Z axis random input	P axis table motion	No further tuning	8 tonne frame
50	Z axis random input	R axis table motion	No further tuning	8 tonne frame
51	Z axis random input	W axis table motion	No further tuning	8 tonne frame
52	X axis random input	X axis table motion	No further tuning	8 tonne frame
53	X axis random input	Y axis table motion	No further tuning	8 tonne frame
54	X axis random input	Z axis table motion	No further tuning	8 tonne frame
55	X axis random input	P axis table motion	No further tuning	8 tonne frame
56	X axis random input	R axis table motion	No further tuning	8 tonne frame
57	X axis random input	W axis table motion	No further tuning	8 tonne frame
58	Y axis random input	Y specimen motion	No further tuning	8 tonne frame
59	Y axis random input	X specimen motion	No further tuning	8 tonne frame
60	X axis random input	X specimen motion	No further tuning	8 tonne frame
61	X axis random input	Y specimen motion	No further tuning	8 tonne frame

The full series of time history test results for the LNEC table have already been published (LNEC, 1996), but a summary of the tests performed is included here (table A.8). The test number is listed along with details of the type of input signal being used, the number of axes being shaken, and the loading on the table.

Table A.8 Time history tests performed at LNEC (Feb. 1996)

Test Number	Type of signal input	Axis being excited	Table Loading
1	Random white noise	Longitudinal	Bare table
2	Random white noise	Lateral	Bare table
3	Random white noise	Vertical	Bare table
4	Random white noise	Longitudinal	8 tonne frame
5	Random white noise	Lateral	8 tonne frame
6	Random white noise	Vertical	8 tonne frame
7	Flexible Payload	All translational axes	Bare table
8	Flexible Payload	All translational axes	8 tonne frame

A.5 Further tests at the Bristol laboratory

A summary of the additional frequency response tests performed at the Bristol laboratory to further investigate some of the issues raised by the earlier tests at the four sites is shown in table A.9. The test numbers are listed, along with details of the type of input, the test type (external accelerometers on the table were used for all the measurements), and the loading on the table.

Table A.9 List of further random signal tests performed at Bristol (July 1997)

Test No.	Axis being driven and signal type being used	Axis being recorded and which instruments being used	Table loading
1	Z axis random input	Z axis table motion (external accels)	Bare table
2	Z axis random input	Pitch_1 axis table motion (external accels)	Bare table
3	Z axis random input	Pitch_2 axis table motion (external accels)	Bare table
4	Z axis random input	Z axis table motion (above Z4 actuator)	Bare table
5	Z axis random input	Z axis table motion (below upper bearing - Z4 actuator)	Bare table
6	Z axis random input	Z axis table motion (above lower bearing - Z4 actuator)	Bare table
7	Z axis random input	Z axis table motion (below Z4 actuator - on reaction mass)	Bare table
8	X axis random input	X axis table motion (external accels)	Bare table
9	X axis random input	Z axis table motion (external accels)	Bare table
10	X axis random input	Pitch_1 axis table motion (external accels)	Bare table
11	X axis random input	Pitch_2 axis table motion (external accels)	Bare table
12	X axis random input	Z axis table motion (above Z4 actuator)	Bare table
13	X axis random input	Z axis table motion (below upper bearing - Z4 actuator)	Bare table
14	X axis random input	Z axis table motion (above lower bearing - Z4 actuator)	Bare table
15	X axis random input	Z axis table motion (below Z4 actuator - on reaction mass)	Bare table

Test No.	Axis being driven and signal type being used	Axis being recorded and which instruments being used	Table loading
16	Z axis random input	Z axis table motion (below upper bearing - Z4 actuator)	5 tonne frame
17	Z axis random input	Z axis table motion (above lower bearing - Z4 actuator)	5 tonne frame
18	Z axis random input (preload press. high)	Z axis table motion (below upper bearing - Z4 actuator)	5 tonne frame
19	Z axis random input (preload press. high)	Z axis table motion (above lower bearing - Z4 actuator)	5 tonne frame
20	X axis random input	X axis table motion (external accels)	5 tonne frame
21	X axis random input	Z axis table motion (external accels)	5 tonne frame
22	X axis random input	Pitch_1 axis table motion (external accels)	5 tonne frame
23	X axis random input (preload press. high)	Pitch_1 axis table motion (external accels)	5 tonne frame
24	X axis random input (table position low)	Pitch_1 axis table motion (external accels)	5 tonne frame
25	X axis random input (table position high)	Pitch_1 axis table motion (external accels)	5 tonne frame
26	X axis random input (table position high & preload high)	Pitch_1 axis table motion (external accels)	5 tonne frame
27	X axis random input - No MCS	X axis table motion (external accels)	5 tonne frame
28	X axis random input - MCS running + composite filters	X axis table motion (external accels)	5 tonne frame
29	X axis random input - MCS running, no composite filters	X axis table motion (external accels)	5 tonne frame
30	X axis random input - No MCS, freq range 0-100 Hz	X axis table motion (external accels)	5 tonne frame
31	X axis random input - No MCS, freq range 0-20 Hz	X axis table motion (external accels)	5 tonne frame
32	X axis random input - MCS running, freq range 0-20 Hz	X axis table motion (external accels)	5 tonne frame
33	X axis random input - MCS running, no composite filters, freq range 0-20 Hz	X axis table motion (external accels)	5 tonne frame
34	X axis sine sweep input - MCS running, no composite filters, freq range 0-20 Hz	X axis table motion (external accels)	5 tonne frame
35	X axis sine sweep input - No MCS, freq range 0-20 Hz	X axis table motion (external accels)	5 tonne frame
36	X axis sine sweep input - No MCS, freq range 0-5 Hz	X axis table motion (external accels)	5 tonne frame
37	X axis sine sweep input - MCS running, no composite filters, freq range 0-5 Hz	X axis table motion (external accels)	5 tonne frame

A summary of the additional time history matching tests performed at the Bristol laboratory to further investigate some of the issues raised by the earlier tests at the four sites is shown in table A.10. The seismic signal being matched is listed, along with details of the loading on the table, the test reference, the type of signal used at the start of the matching and the type of MCS control being used.

Table A.10 List of further time history tests performed at Bristol (July 1997)

Signal	**Table Loading**	**Test number**	**Initial signal used**	**Condition of MCS**
EL Centro	Bare table	Run 1	Displacement	No MCS
EL Centro	Bare table	Run 2	Displacement	MCS - no pre adaption
EL Centro	Bare table	Run 3	Displacement	MCS - gains held constant
Kalamata	Bare table	Run 4	Displacement	No MCS
Kalamata	Bare table	Run 5	Displacement	MCS - no pre adaption
Kalamata	Bare table	Run 6	Displacement	MCS - gains held constant
Kalamata	Bare table	Run 7	Displacement	MCS - no pre adaption, quicker response in Z
Kalamata	Bare table	Run 8	Displacement	MCS - no pre adaption, quicker response all actuators
EL Centro	5 tonne frame	Run 9	Displacement	No MCS
EL Centro	5 tonne frame	Run 10	Displacement	MCS - no pre adaption, composite filters off
EL Centro	5 tonne frame	Run 11	Displacement	MCS - continue adaption, composite filters off
EL Centro	5 tonne frame	Run 12	Displacement	MCS - continue adaption, composite filters off
EL Centro	5 tonne frame	Run 13	Displacement	No MCS - preload high
EL Centro	5 tonne frame	Run 14	Displacement	No MCS - preload high
EL Centro	5 tonne frame	Run 15	Displacement	MCS - no pre adaption, composite filters off, preload high
EL Centro	5 tonne frame	Run 16	Displacement	MCS - no pre adaption, composite filters off, preload high, ΔP adjusted
EL Centro	5 tonne frame	Run 17	Displacement	MCS - continue adaption, composite filters off, preload high
EL Centro	5 tonne frame	Run 18	Displacement	MCS - no pre adaption, composite filters off, preload low
EL Centro	5 tonne frame	Run 19	Displacement	MCS - continue adaption, composite filters off, preload low
EL Centro	5 tonne frame	Run 20	Displacement	MCS - no pre adaption, composite filters off, table low
EL Centro	5 tonne frame	Run 21	Displacement	MCS - continue adaption, composite filters off, table low
EL Centro	5 tonne frame	Run 22	Displacement	MCS - no pre adaption, composite filters off, table high
EL Centro	5 tonne frame	Run 23	Displacement	MCS - continue adaption, composite filters off, table high
Kalamata	5 tonne frame	Run 24	Displacement	No MCS
Kalamata	5 tonne frame	Run 25	Displacement	MCS - no pre adaption
Kalamata	5 tonne frame	Run 26	Displacement	MCS - continue adaption
Kalamata	5 tonne frame	Run 27	Displacement	MCS - continue adaption

Appendix B

Data Processing with the R9211C spectrum analyser

B.1 Exploratory data analysis

The R9211C spectrum analyser used throughout the majority of this research contains a curve fitting algorithm that fits a function either of the form

$$\text{System gain } \times \frac{(s-z_1)(s-z_2)\ldots(s-z_n)}{(s-p_1)(s-p_2)\ldots(s-p_m)} \quad \text{(B.1)}$$

or of the form

$$\text{System gain } \times \left\{ s^{n-m} + L_1 \times s^{n-m-1} \ldots + L_{n-m} + \frac{r_1}{(s-p_1)} + \frac{r_2}{(s-p_2)} + \ldots + \frac{r_m}{(s-p_m)} \right\} \ldots\ldots\ldots\text{(B.2)}$$

to frequency response function (FRF) data, where z and p are complex 'zeroes' and 'poles' of the FRF in the Laplace domain with L and r the complex residues.

The algorithm is not specific to modal analysis of structures driven by point or distributed loads for which case the ideal FRF between at j and force at j for a set of N modes of vibration is:

$$\mathrm{H_{jk}} = \sum_{r=1}^{N} \frac{{}^{r}\varphi_j \, {}^{r}\varphi_k}{-\omega^2 + \omega_r^2 + 2i\zeta_r \omega\omega_r} = \sum_{r=1}^{N} \left(\frac{{}^{r}\mathrm{H_{jk}}}{i\omega - \gamma_r} + \frac{{}^{r}H_{jk}^{*}}{i\omega - \gamma_r^{*}} \right) \ldots\ldots\ldots\text{(B.3)}$$

where $\gamma_r = v_r + i\mu_r$, * indicates conjugate, $v_r = -\zeta_r \omega_r$, $\mu_r = \omega_r \sqrt{1-\zeta_r^2}$ ....(B.4)

ζ_r is the fraction of critical damping, ω_r is the natural frequency (rad / sec),

${}^{r}\varphi_j$ and ${}^{r}\varphi_k$ are normalised mode shapes and ${}^{r}H_{jk}$ is the residue.

The R9211C curve fit provides either poles and zeroes (as here) or poles and residues. From the similar forms of eqn. B.2 and eqn. B.3 the poles provide the natural frequency and damping ratios in the usual form, except that the pole values given are as Hertz and not radians per second.

In general the curve fitting will determine zeroes and poles to fit the data, which the R9211C does not expect to be in the specific form of eqn. B.2. Also eqn. B.3 accounts for the frequency range 0 to infinity while the curve fit is determined for a limited frequency range. As a result, there will in general be poles with positive real parts and other artifices to account for the residuals, the different form of eqn. B.3 for base excitation and other extraneous effects of the system, the signal processing and the instrumentation.

Values for natural frequency (f_r) and damping ratio (ζ_r) in the following pages have been derived by equating the poles of the fitted FRF to γ_r (eqn. B.4) as follows:

real part of pole (if negative) $= -\zeta_r . f_r$

imaginary part of pole $= f_r . \sqrt{(1-\zeta_r^2)}$

B.2 Abbreviations used on figures

Figures produced by the R9211C spectrum analyser use the following abbreviations:

<Coh>	Coherence between channel a and b
<Hab>	Transfer function between channel a and b
LnMg	Amplitude being displayed in linear units
dBMg	Amplitude being displayed in dB units
dB	Units in Decibels
Cs	Cursor values - normally amplitude and frequency are shown
125.0m	Equivalent to 0.125 i.e. m stands for milli
AVG(SUM)	Arithmetic average being used to smooth the data
AVG(EXP)	Exponential average being used to smooth the data (weights the most recently aquired data highest)

Glossary

DOF	Degree Of Freedom
ECOEST	European Consortium Of Earthquake Shaking Tables
EERC	The Earthquake Engineering Research Centre, University of Bristol, UK
FFT	Fast Fourier Transform
FRF	Frequency Response Function (or transfer function)
IFFT	Inverse Fast Fourier Transform
ISMES	Structural Dynamic Testing Laboratory, Seriate, Italy
LEE	Laboratory for Earthquake Engineering
LNEC	Laboratorió Nacional De Engenharia Civil, Lisbon, Portugal
MCS	Minimal Control Synthesis
MDOF	Multiple Degree Of Freedom
NTUA	National Technical University of Athens, Greece
PSD test	PSeudoDynamic test
QA	Quality Assurance
RRS	Required Response Spectra
SDOF	Single Degree Of Freedom
TRS	Test Response Spectra

shaking table	The entire system that is used to provide base excitation testing of structures
(shaking table) platform	The moving part of a shaking table system to which the test specimen (or model) is attached
prototype	The full-scale structure
model	A reduced size copy of the full-scale structure
linear specimen behaviour	Response of specimen is always directly proportional to the input
non-linear specimen behaviour	Response of specimen is not directly proportional to the input. Caused by plastic response or collapse of the specimen
hardware control system (or inner control loop)	The instruments and any other hardware (analogue or digital) that close the basic feedback loops for controlling the table motion
software control system (or outer control loop)	Any software used to acquire data regarding the table motion and generate new drive signals (for real time control or iterative matching)
tuning	The process of adjusting the feedback terms in the hardware control system to make the transfer function of the shaking table as flat as possible across the operating frequency range of the table
iterative matching	Repeatedly modifying the drive signals and using them to control the table until the actual table motion is a close as possible to the desired table motion
linear iterative matching	Use of a linear formula to calculate revised drive signals
non-linear iterative matching	Use of a non-linear formula to calculate revised drive signals
out-of-real time control	See iterative matching

cont.......

real-time control	Control (such as MCS) that modifies the drive signal for the shaking table at the sampling rate of the acquisition of the feedback signal (usually < 1ms) to compensate for any table-specimen interaction

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